Heating · Piping and Air Conditioning

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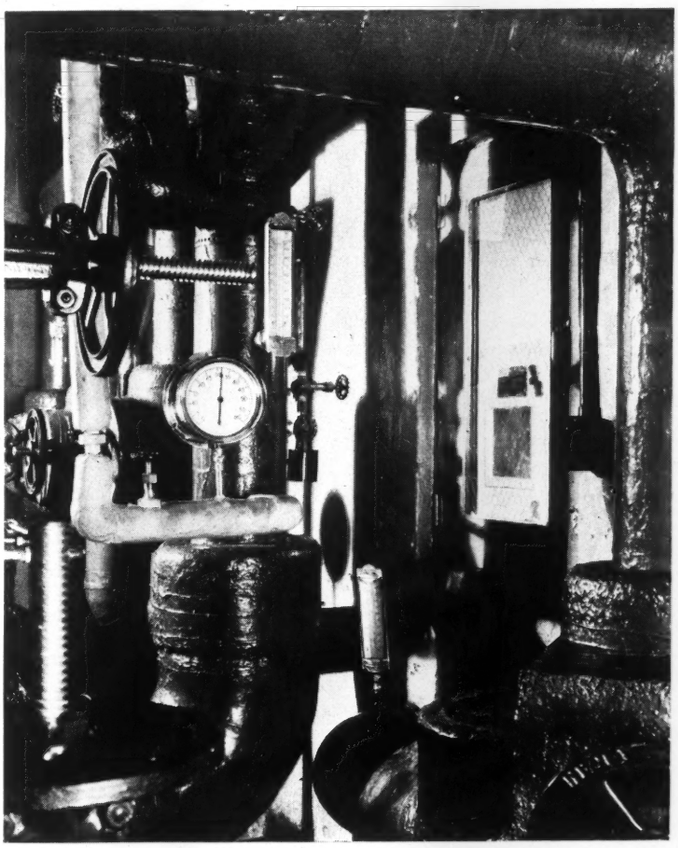
CHOOSING *the right* AIR CONDITIONING SYSTEM

Part 1 - What Is the Proper Viewpoint?

By **A. Warren Canney***



Wendell MacRae.

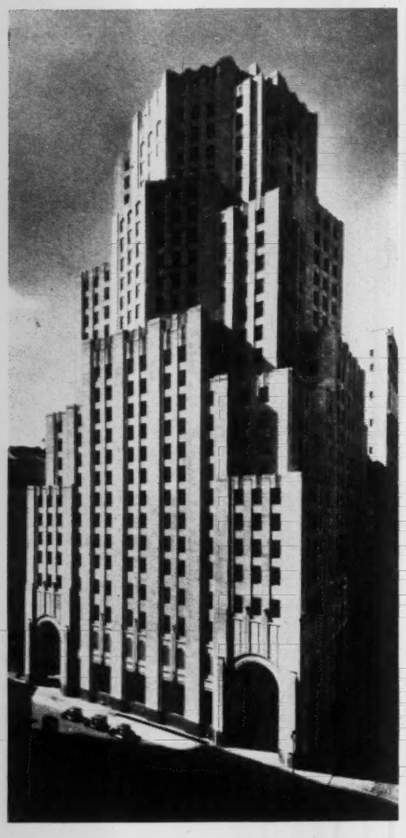
IN THIS series of articles we are going to base our discussion on the main points that are known to influence the choice of an air conditioning method. In too many instances the choice has revolved around minimum requirements (a) to meet price and (b) to meet how little with which a given prospect might be satisfied. The choice is also sometimes inhibited by a limitation of the types or features of equipment that circumstances can allow.

Aside from the ultimate question of cost—which is clearly a matter of careful analysis of each proposed installation, there being no two sufficiently similar—it will have been a worthwhile objective if a foundation is provided for the comparison of methods, of which a wide and highly ramified variety are afforded.

The factors underlying this foundation are five-fold:

1. What a particular installation requires and the precise functions of any equipment to fulfill the requirements.
2. The methods and types of equipments available whereby these functions can be provided for equitable comparison.
3. The suitability of one method over another in meeting the thermal requirements at hand.
4. The suitability of one method over another in meeting the physical requirements at hand.

*The author has completely designed comfort air conditioning systems of all principal types. He completed research on the first full-fledged unit air conditioner for quantity production, designed the first commercial year 'round complete residence air conditioning system. He has designed and supervised the installation and operation of many central station plants, notably those at Rockefeller Center. As an air conditioning specialist of the staff of Clyde R. Place, Consulting Engineer for Rockefeller Center, with Ernest Williams, Manager, he has designed in detail the vast combination system employing units for completely conditioning the 38 story office building now under construction in Rockefeller Center.



Wurts Bros. Photo.

The Metropolitan Life Insurance Company's home office building in New York City is air conditioned throughout by a central station system of air conditioning

It is doubtful if there are any two building air conditioning systems exactly alike; a wide variety of equipment and systems is available, each type having its particular advantages and disadvantages, each with its particular applications, each possible of a number of modifications.

Broadly speaking, air conditioning systems have no generally catalogued classification other than the central station type of systems, unit types of systems and combinations of these two. This classification refers only to the general arrangement of systems with respect to the major apparatus for effecting thermal processes on the air. These main elements for heat transfer are accomplished by various methods. For example, central station systems employ water surface in the form of sprays or metallic surface in the form of coils. In summer comfort conditioning, the coil method may use either direct expansion of the refrigerant or cold water. Either of these sometimes employ an arrangement for flooding the exterior metallic heat transfer surface.

The unit type of system utilizes direct expansion of refrigerant, singly or in groups, with one refrigeration compressor, or a group of

5. The provision of an all-around adaptable and successful installation for the set-up at hand.

Hence, this approach is purely from the purchaser's viewpoint. We are not interested in comparing equipment or apparatus as such. We are highly interested, however in knowing what has solved certain problems and the methods likely to meet the diverse conditions encountered in air conditioning buildings.

The Distinctive Feature of Air Conditioning

The most distinctive feature of air conditioning, the author really believes, is that it is—or can be—the most personal thing scientific achievement ever produced. It can be with you and all around you, touching you constantly, continuously.

Consider any building service or any other article you buy, and figure about how many minutes, hours, or days it is used per year; then compute how much it would cost if it were used as many hours per year as air conditioning. Now compare the two costs. And did you select a personal article for legitimate comparison?

The refrigerator in your home can freeze up or it may unwillingly stop and defrost all over the contents—but nobody *feels* it. Your radio, your car—no physiological upsets occur when they balk at remaining a coordinated mechanism.

In this respect, air conditioning is no different from heating, except that air conditioning is a constant year 'round process—and does much more—when done completely and correctly.

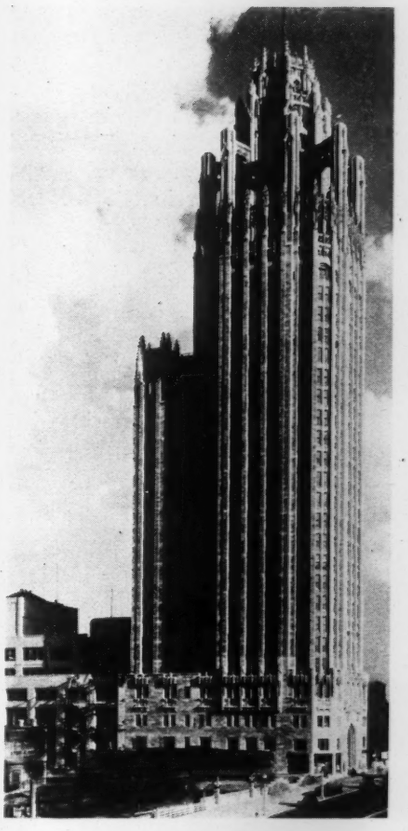
What is there about air conditioning that really warrants all the fuss? Perhaps it's because for millions of years there was no mechanical indoor atmospheric control, heating for a mere several thousand years and only a general demand for 'cooling' during the past decade or so. Its sheer newness, is that it? Perhaps. Let us list some of the major but obvious reasons for the general "excitement" about comfort air conditioning:

1. Air conditioning is supposed to be a new business outlet, born of science and inescapably bound up with things scientific. It has appeal for these reasons alone.
2. Everyone wants always to be comfortable.
3. The new uses of space by modern society, together with highly refined construction methods, otherwise create adverse internal atmospheric conditions which only air conditioning can rectify—hence its timeliness to civilized society.
4. We know it can be a boon to improved health; greater, non-interrupted comfort, increased and better mental as well as physical output. But let us face the fact that we do not yet have either a scientific yardstick nor figures of proved, generally acceptable worth (except in a few isolated and necessarily particularized cases) with which to evaluate all of these advantages. It is, of course, a paying accessory to a great many businesses.
5. Air conditioning requires more space, frequently more architectural consideration, costs more to own and to operate, requires more installation engineering, necessitates more planning in building coordination than any other mechanical equipment for buildings, cubic foot for cubic foot served. Its incorporation in buildings undoubtedly requires more study than any other single mechanical service. We will focus much of our attention on these aspects.
6. Due to the many different types of systems, together with very much simplified methods affording the accomplishment of the same thing, and with new devices and methods always being developed, it is imperative that a greater relative percentage of the total cost of this mechanical building service be applied for skilful engineering. Our attention will be directed to this repeatedly.
7. In that standard objectives are not prescribed for all cases; because such a wide range exists in the physical accomplishments

units may be furnished with cold water from a central refrigeration plant.

The question of the proper method for a particular case excites lively discussion, is important alike to the purchaser of an air conditioning system and to air conditioning engineers. To say that any method is the most suitable is impossible, as every building has its peculiar requirements, each case is influenced by factors not met with in other situations. Each installation requires careful planning with these requirements constantly in mind.

Because of the many sides to the question, it is perhaps overly optimistic even to try to establish a basis upon which the decision can be made. The purpose of this series—which is addressed mainly to the prospective purchaser of a building air conditioning system—is to point out the types of systems which are available to him, and to discuss the factors underlying their selection as a basis for co-operation with air conditioning engineers. This month the proper viewpoint toward air conditioning is considered so that a foundation for the more detailed articles to follow will be had.



Some four hundred individual units were recently installed in the Tribune Tower, Chicago, chilled water being piped throughout the building from a steam jet refrigerating system in the basement

of a system without some owners being much the wiser—dependent on his interest and how long he defers judgment—and because it is still considered the prerogative of insufficiently informed purchasers to lay down the law as to how little they think they are willing to accept in the accomplishments of a system; and last, but not least, despite the "hue and cry" about temperature, humidity and air motion and the splendid researches of the *American Society of Heating and Ventilating Engineers*—the merchandising departments of the air conditioning industry are in many instances split as to what guarantees satisfaction, as to what constitutes minimum technical requirements. This will be clarified.

8. A new and potentially enormous source of revenue for assorted utilities is self-explanatory in the matter of stirring up interest.

Significance of Air Conditioning

We have enumerated a number of matters deserving the attention and serious contemplation of not only the purchasing public, but those who are in any way associated with air conditioning work. We have listed some of the well known high-lights which collectively warrant even more attention than is given to air conditioning today. But much of this space and attention has perhaps missed the mark, because in all the above we have not yet caught the true significance of air conditioning, nor the real reason justifying its widely heralded adoption.

The fact remains that American skyscrapers are universally regarded as man's greatest achievement in building—the culmination of building technique for all time, symbolic of what contemporary science can do for us. But it has been overlooked that most of these functional monuments were built with the million-year-old notion of shelter as a basic objective! Do you realize that with the single exception of heating, these buildings have accomplished little more in the matter of continuous comfort than the first primeval hut? For shelter is the evolutionary basis of all building—and without the advent of air conditioning, it would still be an incomparably crude functional objective. With regard to location, economics, physical refinements, exterior and interior appearances, the labyrinth of mechanical arteries and the intricate sub-divisions which constitute an otherwise modern building—these things have all experienced unparalleled ingenuity and attention. Recognizing that shelter is nothing more than a prerequisite to mere existence and imposing an interpretation that is worthy of an enlightened and intelligent society, the prime function of building becomes *sustained indoor comfort*—which embodies shelter as a rudimentary necessity. This is what architects and builders have been fundamentally wishing for, trying to produce, all the time. But only air conditioning provides the necessary supplementary equipment. The H. G. Wells of a thousand years from now may quite properly say that for the first time in history, the twentieth century saw buildings erected and converted *with the conscious intent* of accomplishing well their very first purpose.

We are now ready to examine what it is that air conditioning should consistently contribute to building.

Considerations in Planning an Air Conditioning System

Just what ought to be done to provide sustained indoor comfort? Is it how little a man wants to pay, or is it what the practical science behind the art should tell

him he must have? While a pertinent question, it can't be immediately settled.

So much emphasis has been placed on temperature, humidity and air motion, that the manner of how these things are effected has been excessively diminished in the purchasers' best interest. 'How' seeks a level of importance with 'what.' A balance must be sought in all that is done. Therein lies the technique of air conditioning.

The following list was evolved as the result of reflection on *all* the main items which should receive balanced consideration. While lists of this sort are common, they have not gone far enough. Its rigidity is subject to experienced discretion according to the variables at hand.

True air conditioning should make provision for effecting the following:

1. *Establish* temperature and relative humidity within suitably narrow toleration limits within the occupied zone of each space conditioned.
2. *Establish* simultaneously, within predetermined limits, not only the rate of air motion but its direction, controlling it between limits without semblance of either a draft or stagnation at any point in the occupied zones.
3. *Establish* a requisite degree of air purity on a specified maximum dust count basis.
4. *Replenishment* with a predetermined minimum quantity of outdoor air to sustain a wholesome atmosphere.
5. *Purge* haze from smoking and other sources.
6. *Prevention* of a detectable concentration of odor vapor.

Note: Unless items 4, 5 and 6 are adequately effected what point can be scored for an investment which considers only items 1, 2 and 3?

7. *To accomplish* items 1 and 2 uniformly in the same space and in adjacent spaces regardless of heat and water vapor gains and/or losses, their sources, or their individually maximum, minimum or instantaneous rates.

These are strictly items of air conditioning but another element has come along, definitely a matter of air conditioning because the physical state of air is involved:

A unit air conditioner in a private office

8. *To scientifically control* sounds generated in any way as a result of operation of the equipment so that the satisfactory, former or existing sound level without the equipment in operation will not be objectionably increased.

Now, dependent on the scope of the system, there are certain additional considerations which must be duly balanced in the incorporation of a system within a building in its entirety or in part.

9. *To provide means* for a sliding scale relation between indoor and outdoor conditions.
10. *To provide instruments* and means for intelligent foolproof operation, cost control and the recording and supervision of results produced.
11. *To provide adequate accessibility* for proper maintenance and quick repairs.

The importance of items *9, 10,* and *11* is incontestable, neglected though they often are. The installation of air conditioning gets a zero rating without crediting it with satisfactory performance obtained reliably and economically. All these things having been achieved, none of the following items can be properly discounted in the planning of air conditioning:

12. *Architectural coordination.*
13. *Operational coordination* with the other mechanical services so as to assure proper operation with a minimum staff.
14. *Coordination* with the other mechanical services, economically as well as physically, balancing electrical and plumbing costs chargeable against the cost of the air conditioning equipment proper.
15. *Establish the total cost* for ownership at a minimum in the location, selection, arrangement of equipment, pumping methods and the like, while not sacrificing any other item herein.
16. *Ascertaining* the rates, demands, loads and cost for utilities without air conditioning and by means of conventional engineering economics establish a total initial and operating cost which provides the economic balance for the building with air conditioning.
17. *Establish the results,* operating method, initial cost, operating costs and all costs chargeable reliably in advance as a basis for investment and budgeting for owner acceptance.

These standards of practice are the results of an evolutionary process toward a higher criterion in air conditioning work. They have been rigorously applied and as written here represent minimum requirements.

It must be remembered that the purchaser of an air conditioning systems gets what he pays for and other folks, Miss Office and John Public, *experiences* what he paid for. There are degrees of results and refinements and one of our essential purposes is to acquaint the reader with them so that he may establish what he should have and what he can sacrifice at his own risk.

Difference Between Industrial and Comfort Air Conditioning

Air conditioning had its commercial origin in factories and its distinguishing feature was the production of a predetermined influence on a material. So skillfully and positively could this be done that the inevitable transition to the comfort air conditioning field brought with it the guarantee of controlling temperature and relative humidity. It was a proved fact that such guarantees provided desirable sales leverage, undoubtedly because of the experienced knowledge gained plus the frequent disappointments encountered without scientific methods or without the improved technique necessary for their positive application. This concentration of attention on

purely psychrometric matters—in a quantitative sense—was deserved. But as a result of factory practice items of vital interest to owners (and therefore architects and consulting engineers) were sacrificed. The total attention to all matters arising by the incorporation of air conditioning in buildings was insufficient.

The following items are common to industrial practice and in their perusal the reader might well contrast them to the requirements of building work:

1. The equipment is exposed for easy access.
2. Appearance and architectural considerations are secondary.
3. The equipments represent a small percentage of total operating costs.
4. Noise and high speeds are usually of no consequence.
5. The maintenance is usually more stringent.
6. Diversified applications call for an alert readiness to provide *any atmospheric condition* rather than *any most suitable method* of providing the same atmospheric requirements as in air conditioning buildings for comfort.

In relation to the numerous objectives previously listed as essentials in the planning of air conditioning, it can readily be seen that any practice set up for industrial work would not necessarily be prepared to strike a correct balance in comfort work.

An Analogy—Consider summer comfort air conditioning. Each room may be likened to a municipal water reservoir, with all the streams feeding it analogous to the many ways heat enters a room. Like reservoirs there are no two rooms exactly alike thermally anywhere in the world!

The great automatic overflow valves, keeping the water level constant, is the air conditioning system, for once installed it regulates the level regardless of which streams pour in water or of any variation in their several rates. Rain may be likened to the people occupying the room intermittently, calling the increase in humidity by evaporation the direct rainfall on the lake, with the streams contributing their heat simultaneously.

The streams which feed the lakes are the walls, floor, ceiling, windows of a room, pouring in heat—more from one than another—and like the stream—the sources may be all on one side, two sides; all individually differing and changing all the time. The problem consists in precisely compensating for the heat and moisture changes that are always taking place—from summer and winter, day to day, one minute to the next.

Due to the many different physical and mechanical processes for so doing, there are many different methods possible for technically bringing about the same result, some much more economical than others, again dependent on many factors surrounding the installation conditions. Moreover, there are as many potential ways to provide air distribution as there is space available for the air conveying ducts. Owners are interested in the method which combines economy and the least obtrusiveness with best technical results.

Thus each installation is in fact technically special if not physically different and there exists a hundred-and-one mechanical arrangements and equipments to accomplish the same approximate result.

In addition there is the elusive psychological element in comfort work, brought about no doubt by people becoming "self-comfort conscious" on feeling or otherwise learning they are in an air conditioned atmosphere. In deliberately making tests along these lines one complaint the author investigated is amusing. The purchaser was kicking; knowing the design figures, the room was overcrowded. A previous check up revealed the system was giving the room 100% capacity. At the end of the time it took the author to smoke a cigarette in a not-distant corridor, he returned to the room and asked the gentleman (with a bright hopeful smile) "How do you feel now?" The question drew a thankful "Fine, thanks" and a cigar in the bargain!

The difference in proportioning of the several ways the body loses heat in summer, the changes automatically effected in the physiological mechanism in its response to liberate moisture at various times with the same person and these differences among people all contribute to make the problem deserving of careful engineering.

How Compare Air Conditioning Systems?

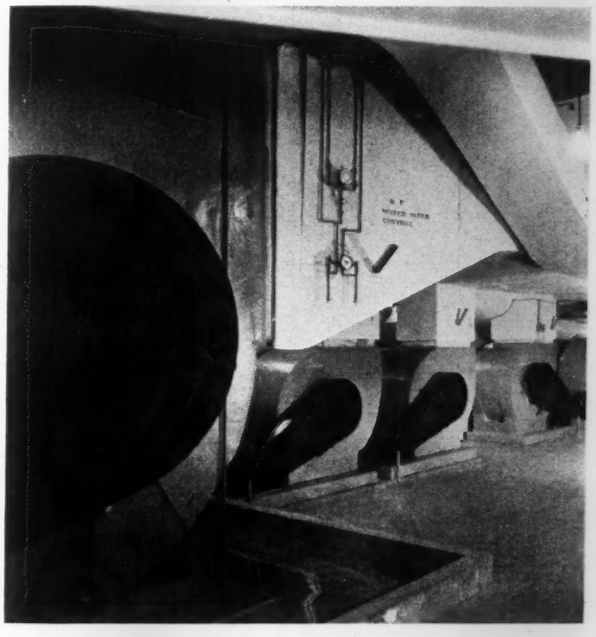
We are faced with the problem of comparing systems which on the one hand require a large crew to operate, with devices on the other hand which receive no more attention than a percolator, yet have a considerably greater amount of mechanism all told with which to do the same work.

There are those who say "why sell a man a Rolls Royce when all he needs is a Ford to give him relief?" Others advocate dehumidification exclusively, others "a little cooling,—sell 'em relief." Others say "there are only two dozen days a year when a man needs it."

We are also faced with products which can legitimately be called air conditioning equipment before they are uncrated, but if one of the vital functions of air conditioning is not effected in the system in which they will become a part, what name should the product have had in the first place?

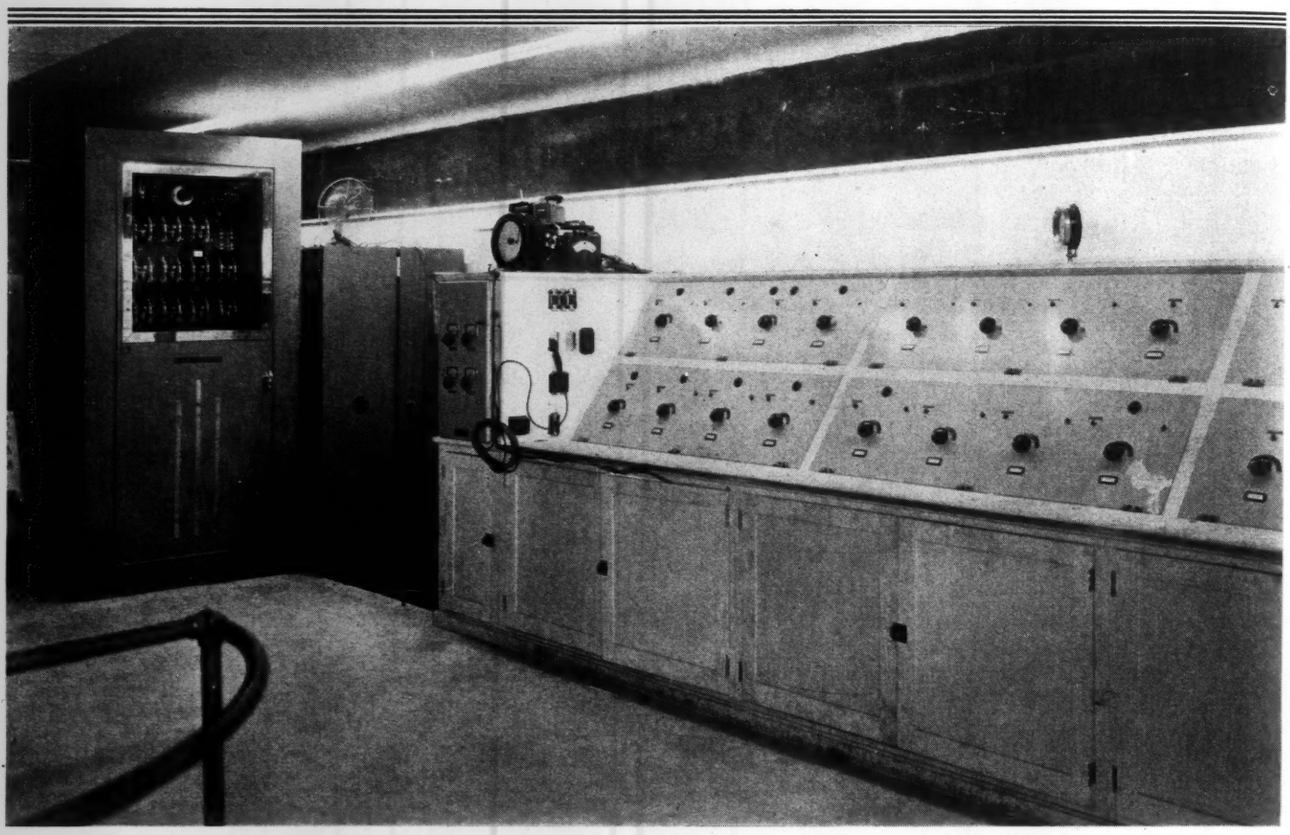
When is a central station system indicated and when

(*Continued on page 75, Back Section*)



Brown Brothers Photo.

A central station air conditioning plant



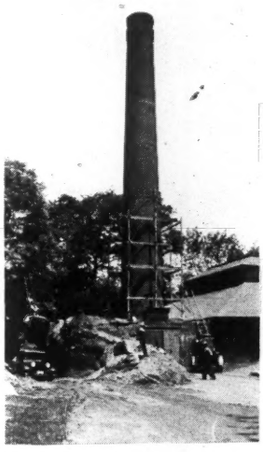
The controls which govern operation of Mount Holyoke heating plant, with the furnace master control on left and building control on right

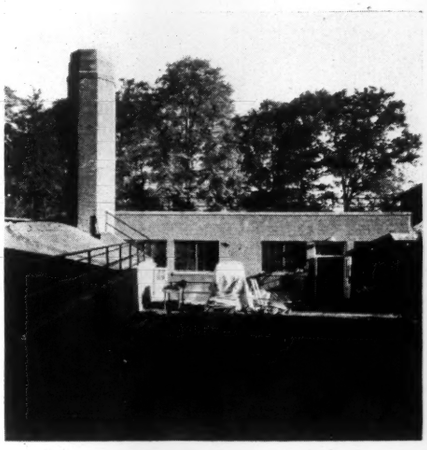
View down an aisle of the central plant, showing some of the 120 oil fired units, which are arranged in 24 batteries of five units each

New Heating Plant of Radical Design

Mount Holyoke College Modernizes Central Heating System — 120 Oil-fired Units Replace 7 Hand-Fired Boilers — Flexible, Economical, Fully Automatic Operation the Aim

By **B. R. Sheldon***
with **F. A. Westbrook**





LEFT TO RIGHT—Old stack and central heating plant, showing excavation for new plant at left of view. . . . South view of new central plant and the new stack. . . . New central plant, the enclosure within the railing being the roof; the roof of the old plant is seen in the rear

MOUNT HOLYOKE COLLEGE at South Hadley, Mass., has just completed the installation of an unusual central heating plant consisting of 120 oil-fired boiler units arranged in 24 groups of five units each. Approximate cost of this modernization project was $180,000 and it is estimated that the investment will be amortized out of savings in a period of eight years.

For several years the College has had a new heating plant under consideration; first, because of the obsolescence of the old one and second, future architectural developments require the central plant site for landscape development. A new central plant location had been tentatively chosen but the expense of the project in whole was prohibitive. It became imperative because of the age of the old boilers to solve the problem in another way. The new plant was then designed to be placed in a part of the old central plant. Space formerly used as coal storage at the rear of the plant was available. The housing of the new plant adjoins the rear of the old plant and is so designed that at any time the old building may be removed, leaving an inconspicuously housed heating plant hardly visible from any point on the campus. The roof of the new plant is of concrete slab construction covered with 2 ft of loam and sodded over. The whole building, with its 30 ft campanile stack, blends nicely into the surrounding grounds, thus fulfilling the architectural requirements.

Economy of operation was also a guiding factor inasmuch as the old system of heating was without temperature control of any kind and inefficient in its production and use of steam. The new plant is extremely flexible due to the multiplicity of units and the method of control. Units will operate at 80 per cent efficiency, and at no time will it be necessary to operate below this point because only sufficient units will be operated to supply the load demand.

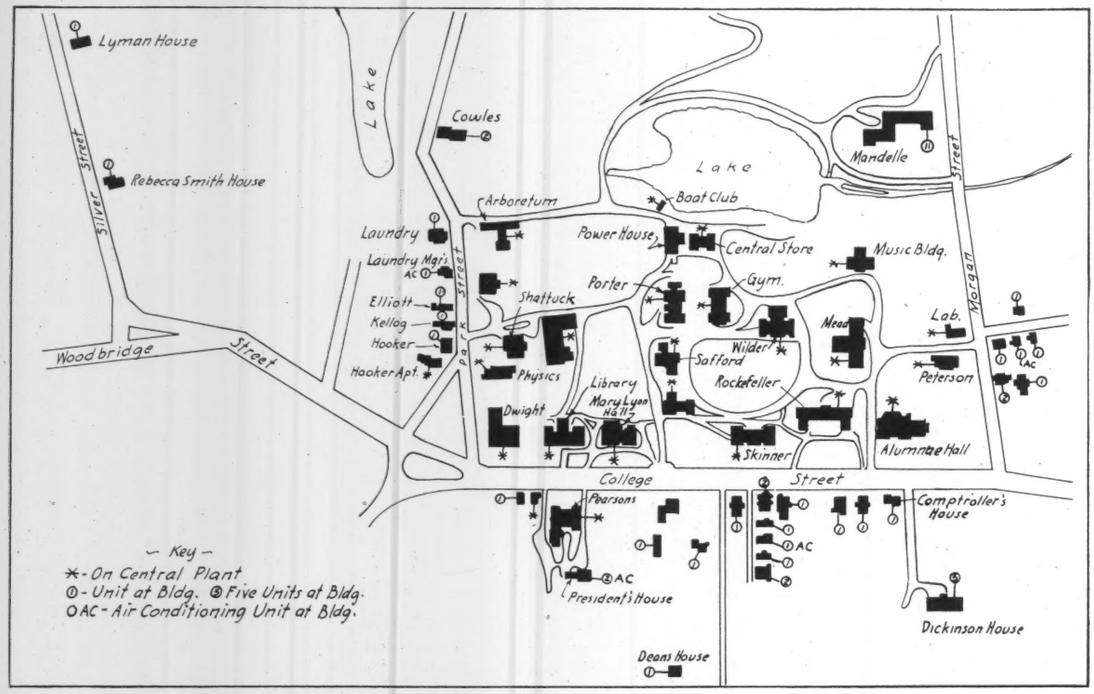
Seven 115 Hp Boilers in Old Plant

In order to appreciate the importance and significance of the new installation it will be worthwhile to consider briefly what the old one it replaces was like. It comprised seven 115 hp H.R.T. boilers, hand fired with bituminous coal and serving approximately 127,000 sq ft of radiation, and heating water for domestic use in the 25 College buildings which have a total tank capacity of 6000 gallons. Most of the building heating systems are one-pipe.

*Chief Engineer, Mount Holyoke College, South Hadley, Mass.

Credit for the design of the plant is due to C. W. Colby, Consulting Engineer for the General Electric Company collaborating with Hollis French Company for the College. The building heating control system was developed by Karl W. Rohlin of Warren Webster Company.

About 17 per cent of the power consumption at the College was generated by means of two steam engines, one rated at 75 and the other at 150 hp. Steam was generated at 75-lb pressure and exhausted to the heating system at 5 to 10 lb. Makeup steam was added at the plant with no pressure reduction at the buildings. The connected radiation (which is retained) is entirely on the campus with the exception of three buildings, all being



This map indicates the College buildings which are heated with steam from the central plant, and those which have individual units at the building

supplied through underground distribution lines. These lines cover a radius of approximately ¼ mile, the longest run being about 2000 ft. Main and return lines are run in tile conduit insulated in the usual manner. Distribution lines from the central plant form a loop system about the campus.

Sizes of steam mains underground vary between 4 and 8 in. and return lines (which are in the same conduit) between 1½ and 3½ in. The early installation was designed for high pressure distribution with pressure reduction at the buildings. This necessitated the use of a third high pressure drip line which will be discontinued with a considerable saving in heat losses.

"Off campus" houses were formerly heated by individual steam or warm air systems burning anthracite. The load for these buildings is equivalent to about 26,000 sq ft of radiation.

The New Plant

For the new system the seven H.R.T. boilers were replaced by 120 oil fired boiler units set in 24 batteries of five each. Rated output of each unit is 275,000 Btu per hr, making a total of 1,375,000 Btu per hr for each group of five. The arrangement is shown schematically in one of the diagrams. The 24 groups of five units each are all connected to a main steam header.

The operation of the entire plant is completely automatic, a centralized electrically operated master control board governing the sequence of operation. A step regulator transmits variations of pressure to a timing device which in turn actuates the necessary electrical relays to start and stop the units, which operate in the groups of five. Inter-connected with the master control board is the induced draft control and combustion air fans. As the number of groups in operation increases, additional draft and air are supplied by increasing the speed of the induced draft fans and starting combustion air fans.

Constant draft conditions are maintained by changing the speed on the induced draft fan motor. To care for intermediate variations between speeds a diaphragm control is used; this control has electrical contacts which actuate a motor driven blast gate in the main flue.

Operating air temperature will not exceed 90 F. Air is delivered to the operating floor and removed from the floor below. The induced draft fans (which draw away the products of combustion) are in duplicate and provided with centrifugal switches which automatically cut out a disabled fan and put the second into service; pilot lights show the operating conditions of the fans. Should both fans fail the plant is protected against further operation.

Combustion gases are collected into a central duct system and discharged at high velocity through the 30 ft stack, which replaces the old 98 ft stack.

A panel board carrying 144 pilot lights arranged in groups of six lights each indicate with the yellow center light that a group of five units is called into service. The five green lights about the center light indicate each individual unit in the group as it comes into operation. A single toggle switch on this board will shut down the entire plant. This panel board duplicates, in lights, the positions of each group and unit on the floor.

(*Continued on page 72, Back Section*)

INDUSTRIAL PROCESS PIPING—

Welding and Brazing Joints and Connections of Industrial Vessels

Welding has been widely adopted for industrial process piping, the proper methods being well understood—more so than are the problems met with in making connections to vessels and containers. The latter are so varied in type of construction that it is difficult to formulate any standard practice except through wide experience spread over the needs of many plants. This article is based upon experience of this kind.

By Lee. P. Hynes* *and* W. Schoeni†

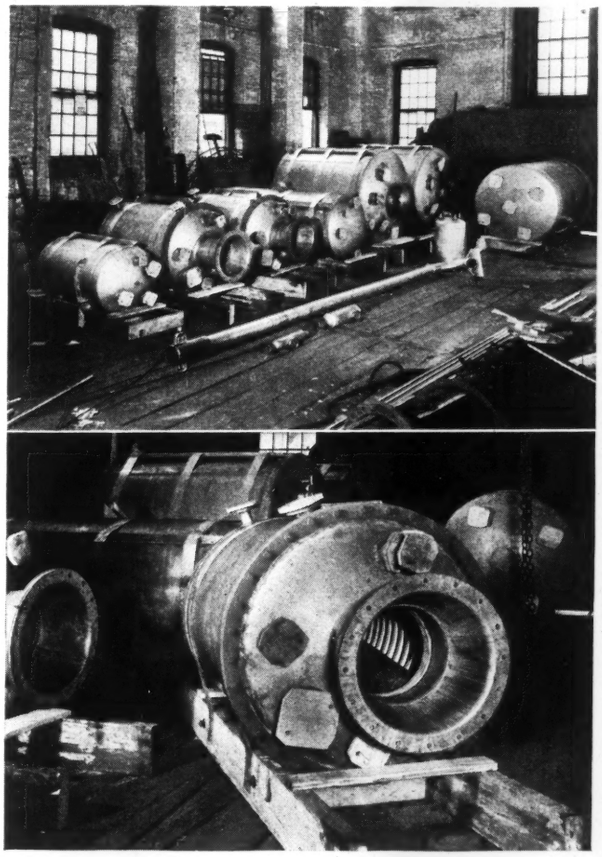


Fig. 1—A group of welded vessels skidded for shipment to a chemical plant. The pipe outlets are protected with blocks of wood

Fig. 2—Close-up of a welded aluminum pressure vessel with an internal heating coil. A welded on, flanged, stuffing box is shown around the pipe connection in the removable head. The coils are made continuous by butt welding sections of tubing

IT IS not our purpose here to present a treatise on the technique of welding and brazing metals, but rather to bring to the attention of engineers and draftsmen some important details in the practical design of process tanks, vessels, and their connections where welding or brazing is required in their construction or erection. The advantages and disadvantages of different types of joints for various metals and under various service conditions will be pointed out.

Only too frequently an otherwise excellent design of container is weakened for its particular service or the cost of construction is unnecessarily increased by a faulty joint design or welding specification. While it is not possible to discuss in this article the design of every type of industrial vessel, the information given has been proved by experience and is believed to represent most reliable practice for general use in work of this character.

Materials commonly used in constructing industrial process vessels include steel, monel, aluminum, nickel, copper, brass, stainless steel, and everdur. Important factors in selecting a particular material for a given job are its cost and its strength and resistance to corrosion in the service for which it is intended. It is possible to weld all of the materials mentioned, but different materials require different types of joints for best results.

In selecting the type of welding for a particular joint there are six principal factors to consider—mechanical strength; uniform density of weld structure to resist leaks; resistance of the weld to corrosion; cost of the welding process; thickness of metal to be joined; and danger of warping or harming the structure of the metal by overheating.

Recommendations for Welding Practice

Steel—Steel may be welded either by the electric arc or by gas. It is the authors' experience that gas welding is usually preferable for thin sheets and the joint is less likely to leak from being porous. For sections above $\frac{1}{8}$-in. thick, electric arc welding is usually cheaper and equally satisfactory in quality. A properly made weld

*Consulting Engineer and Member of Board of Consulting and Contributing Editors.
†Jos. Oat & Sons, Philadelphia.

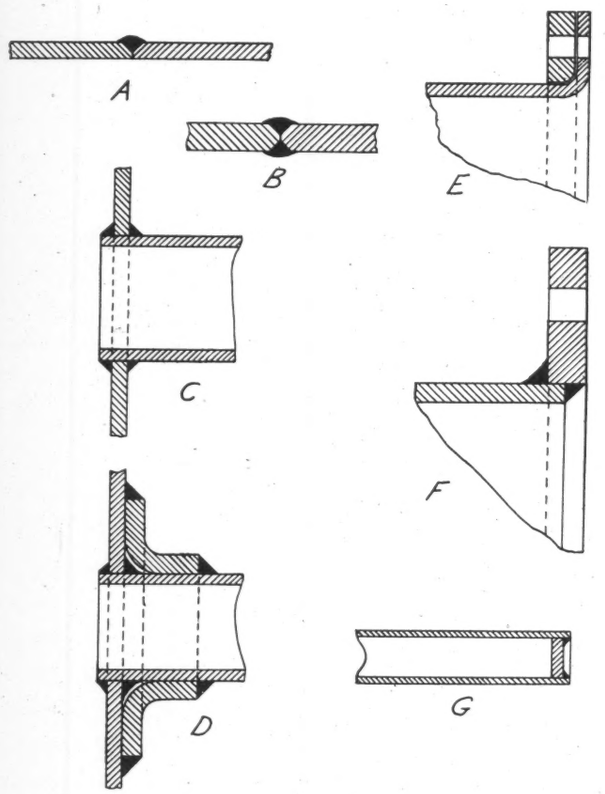
is as strong as the sheets themselves. The electric arc weld employs a more localized heating and causes less warping than gas welding. In welding together plates of about ¼-in. in thickness and upward, the edges should be beveled as illustrated in sketch *A* and for thick plates, especially where subject to heavy strains, the seam should be welded on both sides wherever possible as shown in sketch *B*.

Nickel and Monel—Nickel and monel can be welded very satisfactorily by the electric arc where strength of weld and resistance to leaks are the factors to be considered. However, the filler material used in welding is attacked by some acids and chemicals which do not affect the sheets themselves, hence the electric weld may fail from corrosion. This is not true of gas welding where filler rod material is used of the same composition as the sheets being welded. In this case the weld has the same resistance to corrosion as the sheets. Where nickel or monel castings are welded, the electric arc is preferable, wherever it can be used, because the gas flame tends to overheat and cause small fracture lines in the castings. When welding nickel sheets, the heating causes a change in the crystalline structure on each side of the weld, forming large crystals which are plainly visible; but this does not have any harmful effect upon either the strength of the plate or its resistance to corrosion.

Aluminum—Aluminum should be gas welded, using the same material as the sheet for filler rod. A very excellent weld can be made which is fully as strong as the balance of the sheet and is equally resistant to corrosion.

Stainless Steel—Early attempts to weld stainless steel were unsatisfactory but the difficulties have been overcome by improvements in the composition. Material having approximately 18 per cent chrome, 8 per cent nickel, and low carbon content can be very successfully welded without resultant fragile areas occurring in the sheet near the weld. It is important however to keep the welding heat as low as possible to avoid overheating.

(*Continued on second page following*)



A & *B*—A butt weld is an excellent method for joining plates or tubes. For material ¼ in. or thicker the edges should be beveled to give more welding surface, and for heavy sheets where great strains must be endured it is best to bevel and weld both sides of the sheets as shown at *B*.

C—A good way to attach a branch outlet or a pipe is to extend the end through the vessel wall and weld it securely. The welding should be done on both sides where accessible.

D—Where a branch or pipe is subjected to great strain, a heavy forged or cast reinforcing saddle flange should be applied as shown.

E—The full flanged joint is a very satisfactory method for joining together pipes and cylinders or for applying heads to cylinders. This is particularly true for copper, aluminum, etc. The metal itself is flanged out as shown with a heavy steel backing ring and through bolts to draw down upon a suitable gasket.

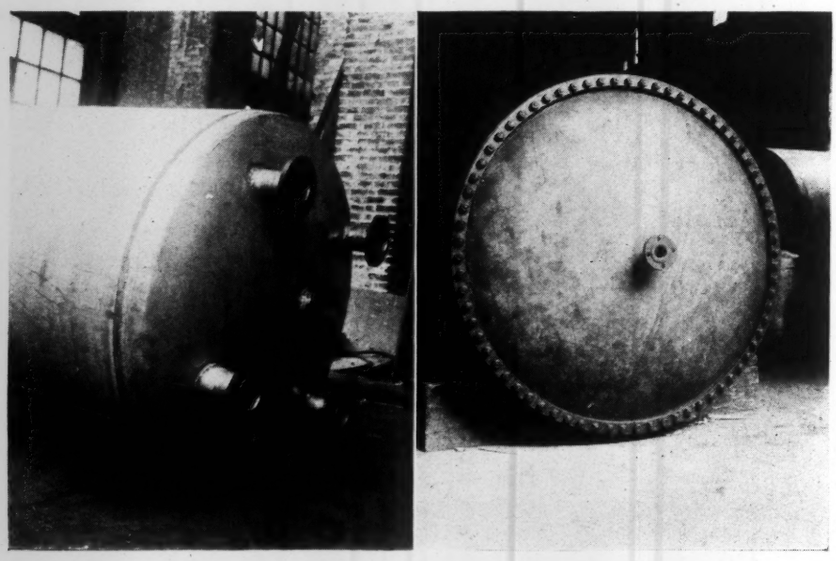
F—Where the material of the cylinder is too heavy for satisfactory flanging or too wide a flange is required, the best method is to fit a heavy ring over the outside of the cylinder as shown and weld it securely on both sides. This is a common method in steel work.

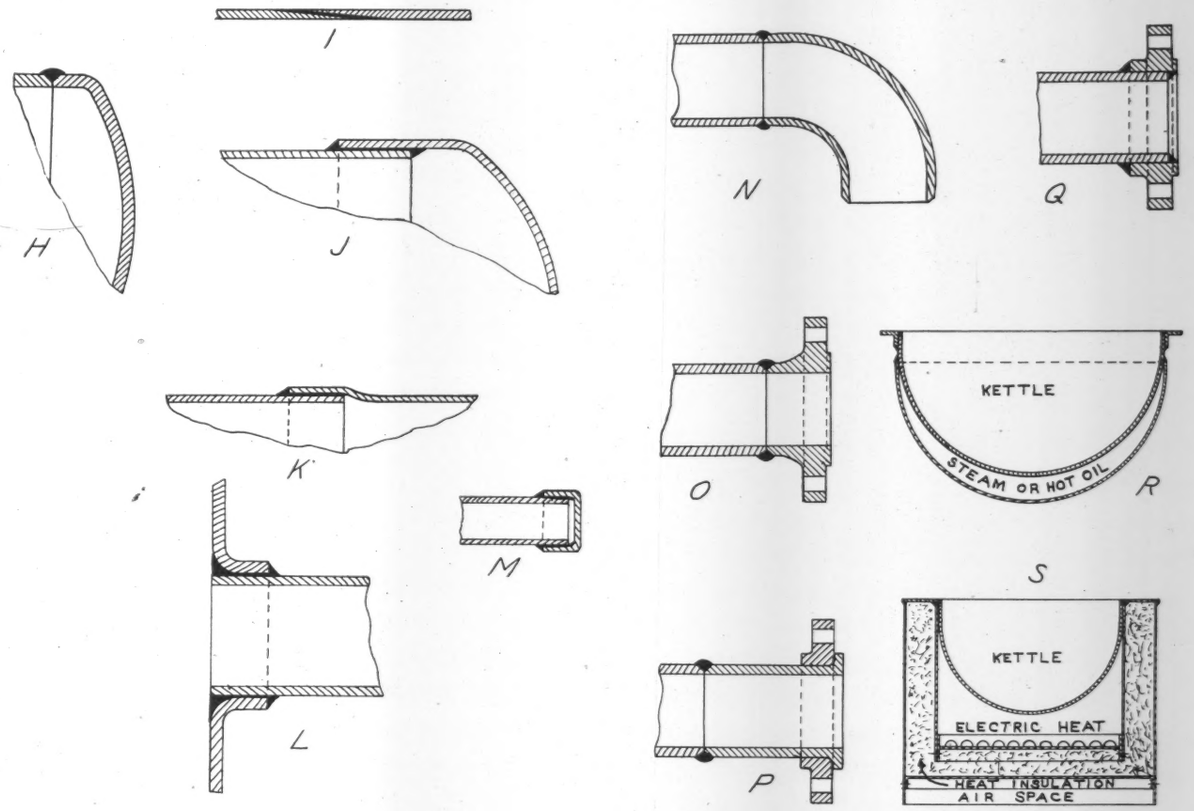
BELOW—LEFT

Fig. 3—A large welded pressure vessel with pipe outlets having Van Stone flange connections. Note the butt weld for the convex head (see drawing H), and the same type for the longitudinal seam (see drawing A). Fig. 4 shows the other end of this tank

RIGHT

Fig. 4—A stainless steel convex head attached to tank shown in Fig. 3 by a full flanged joint 6 ft in diameter. In the center is a small pipe connection with a Van Stone flange





G—The best method of capping a pipe or tube is to force a metal disc inside of it and then weld the joint as shown. Sometimes as an additional precaution the tube end is rolled over before welding.

H—One of the best ways to attach a convex head to a vessel is a butt weld. Large vessels usually have a manhole which permits welding both sides, as shown in *B*.

I—Where brazing or silver soldering is to be employed a butt joint is not satisfactory. The strength of the joint depends upon the brazing material flowing between the pieces being joined. Plates can best be joined by lapping their edges with a long taper as shown. This gives the large surfaces which are necessary for a strong joint.

J—The correct joint for a brazed head on a pressure vessel requires the head to be fitted outside of the cylinder and brazed as shown. It is a weaker construction to put the head inside of the cylinder because the pressure tends to force the cylinder outward away from the head and the pressure on the head tends to tip the flange of the head inward away from the cylinder—thus the pressure tends to disrupt the joint. With an outside head as shown, the pressure tends to force the cylinder out against the head and the pressure on the head tends to tip the flange inward against the cylinder so that the tendency is to press the joint together instead of forcing it apart. This is an important point which is sometimes overlooked in designing pressure vessels.

K—A desirable method of making a brazed joint between pipes, tubes, or cylinders is to slightly expand the end of one section so that it will slip over the next section as shown with just enough clearance for the brazing material to flow freely into the joint. If a smooth surface is wanted, a tapered lap joint, as in *I*, is satisfactory.

L—When making a branch outlet or brazing a pipe to a sheet, a good method is to cup the sheet and fit the tube inside as shown. This strengthens the joint and gives ample surface for brazing.

M—A strong way to close the end of a pipe or tube by brazing is to fit a cap over it.

N—Welding elbows are available in most standard pipe sizes and in many materials. The ends are circular and true to size and come beveled, ready for butt welding to pipe of corresponding size.

O—Welding neck flanges can be procured with the end beveled for butt welding, as shown.

P—Flanged welding nipples with loose bolt flanges can be purchased with the ends beveled for welding. The flanged face of the nipple is finished to provide a good gasket surface. The loose rings facilitate lining up the bolts in making up a joint. This is the Van Stone type of joint.

Q—This illustrates the slip-on type of flange which fits over the end of the pipe and is welded as shown.

R—Kettle with welded-on exterior heating jacket. Pipe connections for the steam or hot oil circulations are not shown.

S—An electrically heated kettle showing a practical method of arranging heat insulation material between double walls around the electric heating chamber. Fig. 9 shows an electric heater for an installation of this kind. The sectional refractory insulators have reflecting faces which throw the heat upward, over the bottom and sides of the kettle. Radiant heat also impinges upon the kettle. The heat is uniformly distributed by this method and can be easily controlled as desired.

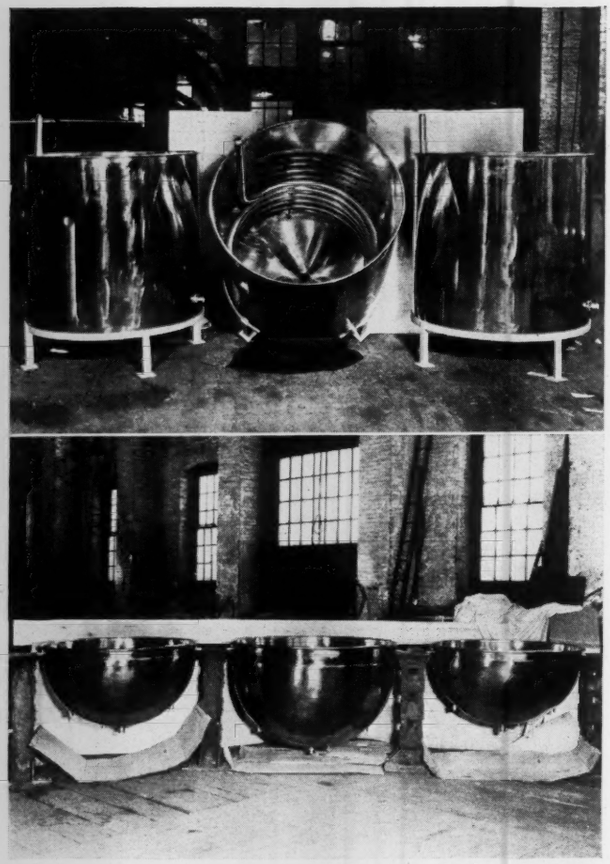


Fig. 5—Welded nickel kettles with heater coils of 2 in. nickel tubing made continuous by butt welded joints (see drawing A)

Fig. 6—Welded stainless steel kettles with welded outer jackets to contain steam or hot oil. (see drawing R)

While stainless steel of the above composition can readily be welded with good mechanical strength, a narrow band on each side of the weld becomes more subject to corrosion than the weld or the sheets themselves. This comprises the area which reaches the critical temperature of between 1200 and 1500 F. For electric welds where the heat is kept low, this corrosion band is very narrow, being merely a line about ¼ in. from the weld. For gas welds this band may be ½ to ¾ in. wide because of higher heat absorption. To overcome this corrosion band adjacent to welds in stainless steel it is necessary carefully to heat-treat the work after welding. Proper heat treating will restore the original resistance to corrosion. Unless this precaution is taken, the full advantages of stainless steel containers will not be obtained and disappointing results may follow from corrosion.

Everdur—Everdur is an alloy of copper, silicon and manganese. It has about the same strength as steel and is very resistant to corrosion. It can be satisfactorily welded by the electric arc or by gas. Welds are strong and resist corrosion the same as the sheets.

Brass and Copper—While brass and copper can be electrically welded, in the authors' experience it is not usually satisfactory to do so. With brass, the zinc content tends to vaporize out. When copper is welded electrically, the weld has the same resistance to corrosion as the copper sheet and will also make a good electrical connection but the weld is likely to be brittle and not generally satisfactory for the construction of pressure vessels.

Brazing is believed to be the most satisfactory joint for brass and copper. It differs from welding in that the filler material is not the same as the sheets and must be "flowed in" between large faces of the joint in somewhat the same way as solder. A properly constructed brazed joint is very strong and free from tendency to leak. It should be remembered however, that the actual joint is sealed by the brazing material. This is usually a composition of brass and for some purposes this material will not resist corrosion as well as the sheets which are joined by it. Consequently it is often necessary to braze the joints of copper vessels with silver or some other filler material which has suitable resistance to the chemical action to which it will be exposed.

The sketches illustrate suitable types of joints for different materials. It is not claimed that these are the only ways in which satisfactory joints can be made but these are well proved methods which can be depended upon and their general use will overcome many of the troubles frequently encountered in work of this character. They represent sound practice from the standpoint of real economy, taking into consideration both first cost of construction and life in service.

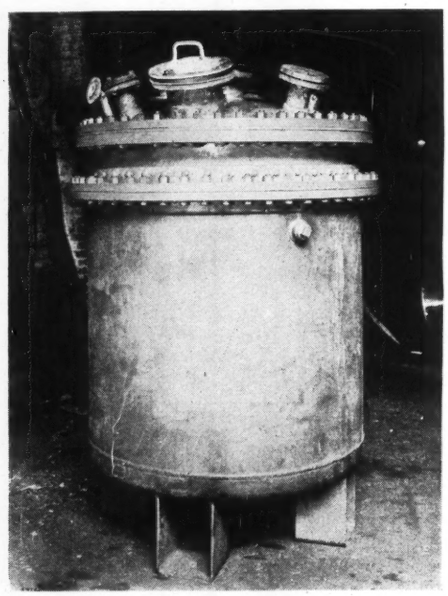
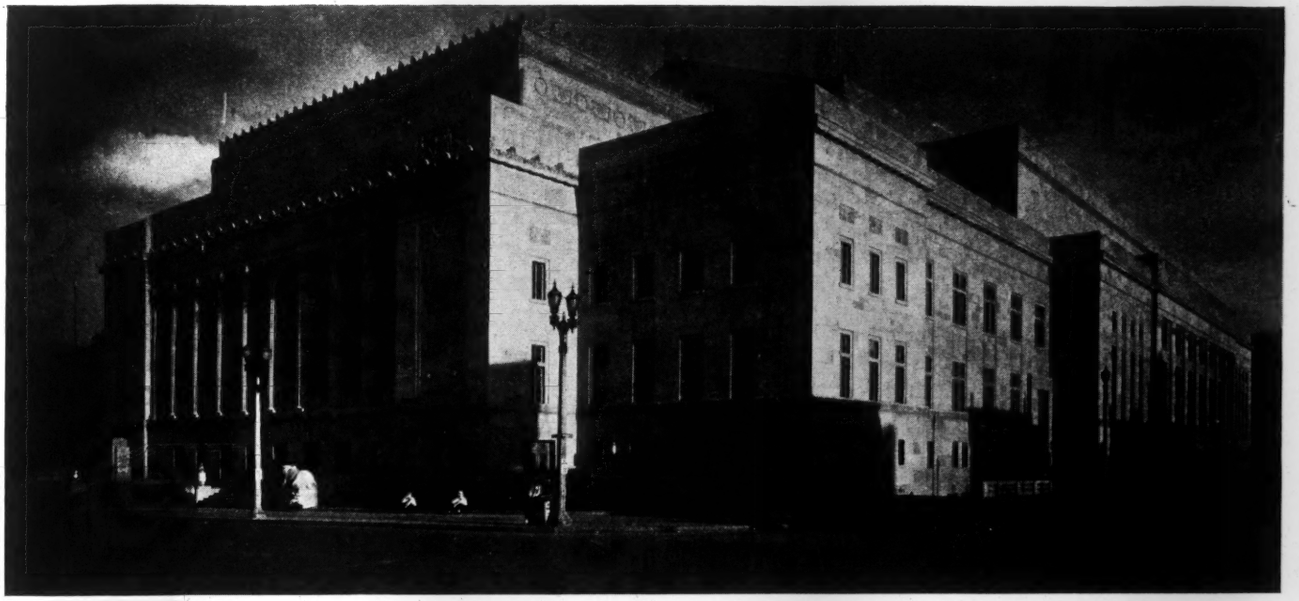


Fig. 7—A copper kettle with a steel outer jacket for steam or hot oil. The outlets in the copper head show the brazed cup joints (see drawing L). The bottom head of the steel heater jacket is attached with a butt weld (see drawing H)

Welding Jacketed Vessels

Often it is necessary to construct vessels having jackets for heating purposes. Sometimes these jackets are arranged to contain steam or circulating hot oil, and in other cases they contain electric heaters. With the first

(*Continued on page 75, back section*)



The St. Louis Municipal Auditorium and Civic Center, planned to provide facilities for meetings of educational societies, musical events, conventions, expositions and other civic enterprises, was completed in April, 1934. The Auditorium enables St. Louis to accommodate both large and small meetings at a convenient location easily accessible from the Union Station and downtown St. Louis.

The illustration indicates the adaptability of the building to various uses. The front entrance at the center of the building on the north side leads to a theater with a seating capacity of 3,500. The side entrances on the north lead to four small assembly halls, two on the east and two on the west side, each with a seating capacity of 750. The rail at the right along the west side of the building protects a stairway that descends to the exposition hall. The canopy at the center along the west side of the building leads to the great convention hall, with a seating capacity of 13,000, and to the exposition hall which occupies practically the entire lower level of the building. The entrances on the east side are the same as the west side entrances. Vehicle entrances are easily accessible for both the convention hall and the exhibition hall. The building measures 466 ft long and 320 ft wide.

All parts of the building can be used at the same time, or any one room may be used separately. Individual dressing rooms, projection booths, stages, elevators and entrances are all arranged for convenient use without interference one with the other.

The architects were The Plaza Commission, Inc., La Beaume & Klein, St. Louis. Engineer, Geo. E. Wells, Inc., St. Louis.

SELECTION OF MOTORS for Air Conditioning

By **Oliver S. Imes***

Selection of the right type of motor for supply and exhaust fans, pumps, air filters and other air conditioning equipment is highly important to economical and flexible operation of the system. The author takes the new St. Louis Municipal Auditorium—which has installed some 74 motors ranging in size from ¼ to 100 horsepower—as an example upon which he bases this informative article on motor selection

MORE than 74 motors, ranging in size from ¼ to 100 hp and totaling approximately 1,000 hp, are installed in the new St. Louis Municipal Auditorium (see illustration and caption) to drive the supply and exhaust fans, dehumidifier pumps, self-cleaning air filters, condensate pumps, and other equipment. Obviously it was highly important to choose the proper motor for each application; the purpose of this article is to explain the selection of motors for air conditioning systems, using this job as an example.

The air conditioning system is so designed that all parts of the building can be served effectively under any condition of use. Adequate air conditioning is, of course, essential to the health and comfort of the great crowds of people using the Auditorium, and this equipment, consisting of exhaust and fresh air fans, air filters, dehumidifying pumps, air washers,

*Engineer, Century Electric Company, St. Louis, Mo.

heating coils, etc., together with the electric motors for driving the various units, comprises the major part of the mechanical equipment of the building.

The Air Conditioning and Heating Systems

A description of the operation of the air conditioning system will indicate some of the considerations involved in motor selection:

Ten large fan rooms and seven additional smaller fan ventilating units control the condition and amount of air supplied to different parts of the building. Four of the large fan rooms supply air to the exposition hall, two large fan rooms supply the theater, one large fan room supplies the two east assembly halls, another large fan room supplies the two west assembly halls and one large fan room supplies the east and one the west dressing rooms. The seven smaller units are exhaust fans that remove the air from rest rooms, halls and places not easily reached by the regular duct system. These smaller units eliminate air from the system that is not desirable to recirculate.

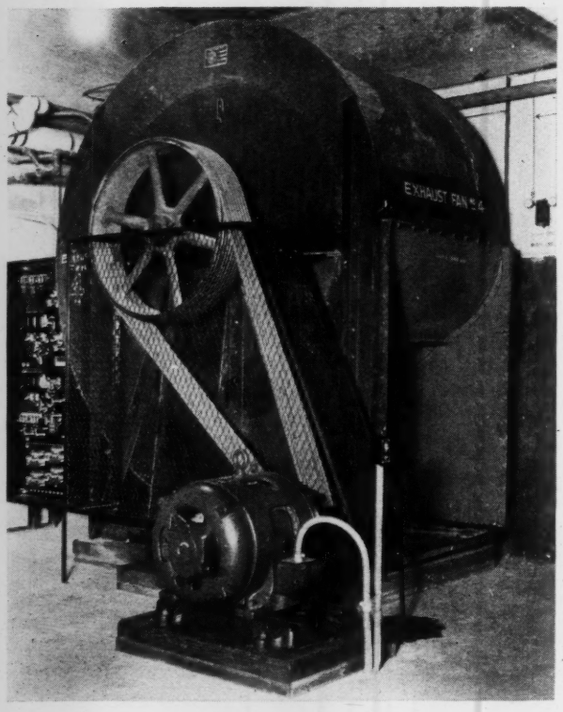
The ten large fan rooms are all alike, except as to size. They all have four motor driven units consisting of an exhaust fan, a pump, a self-cleaning filter and a supply fan. The exhaust fan draws the air out of the ventilated room at the floor level and pulls it toward openings to the outside air and to the recirculated air intake. These openings are of such sizes that a maximum of 75 per cent recirculated air only is possible. The proportion of recirculated air can be reduced as desired until all fresh air is drawn from the outside. The fresh air enters the building, mixes with the recirculated air (if any is being used), and passes through the self-cleaning filter and the heating coils into the air washer and dehumidifier.

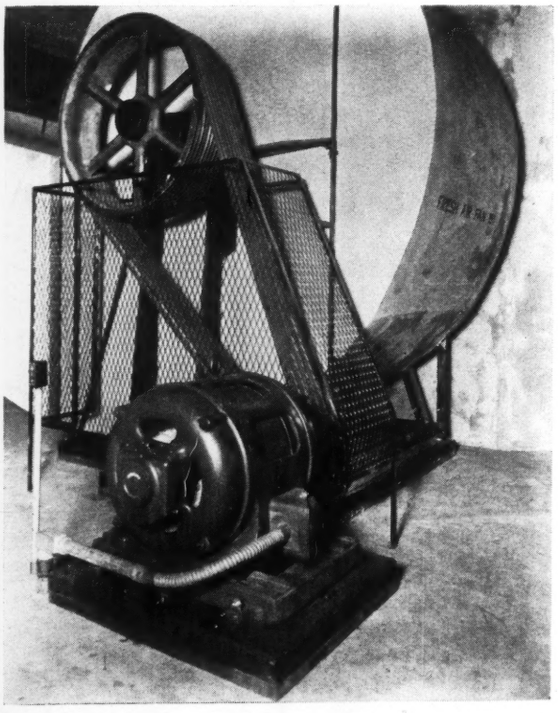
The air washer and dehumidifier is arranged to perform two functions. The centrifugal pump is used to circulate water from the tank under the sprays back through the spray nozzles in an ordinary air washing operation, when it is a question only of cleaning the air. In hot weather a change of valve settings supplies cooled water to the spray nozzles under 30 lb pressure and at 38 F for cooling and dehumidifying.[1] The spray now performs two functions, cooling and washing. The air washer pump now returns the water at approximately 45 F.

From the air washer the air next passes through tempering coils that regulate the temperature of the air entering the room, and from the tempering coils through a supply fan that forces the air through the duct system serving the air conditioned room or rooms. In general, the conditioned air enters the rooms at the ceiling level.

The supply fans are all larger than the exhaust fans, which results in a slight air pressure being maintained in the rooms. Outside air is thus prevented from entering the room, and a uniform condition can be maintained.

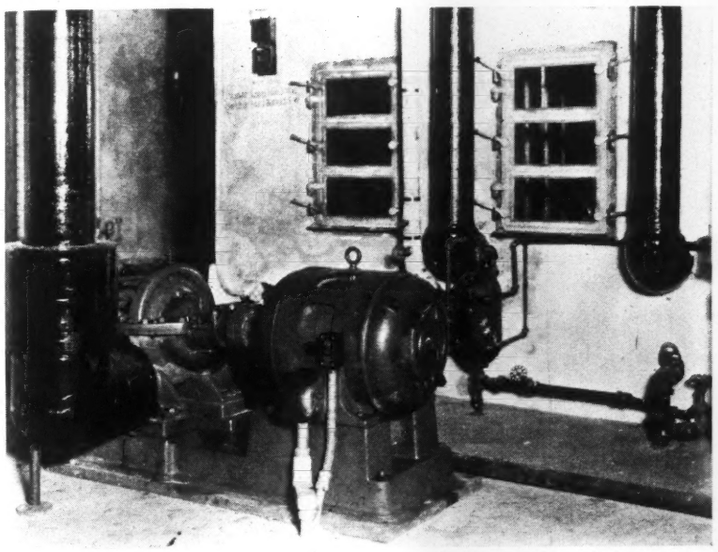
The fan and pump motors, being located in the different fan rooms in different parts of the build-





Lower left—Exhaust fan No. 4, Exposition Hall, with 10/2.5 hp, 870/440 rpm, 60 cycle, 440 volt, two-speed squirrel cage motor. Below—Supply fan in fan room No. 1, with 40/10 hp, 880/430 rpm, 60 cycle, 440 volt, two-speed squirrel cage motor. Right, above—20 hp, 1750 rpm, 60 cycle, 440 volt squirrel cage motor driving pump in No. 4 fan room.

[1]The method of cooling the water has not yet been decided. There are three methods available. (1) From a central plant refrigeration system available to downtown St. Louis; (2) from a new central plant installed in the municipal heating plant and (3) to install refrigerating equipment in the machine room of the Auditorium itself



ing, are arranged for starting or stopping of the motor by push button control; they cannot be started at any other place. This gives the attendant an opportunity to observe whether the equipment is operating properly. The motors are connected for changing speeds or stopping at the main control panel in the main machinery room.

The filter motors operating the self-cleaning filters are interlocked with the pump motor. Thus all the air cleaning equipment for each fan room is started at the same time.

The dampers which control the amount of recirculated and outside air are controlled by compressed air from the central machinery room.

During the heating season temperature is automatically controlled by thermostatically controlled steam valves which regulate the steam entering the heating and tempering coils. Steam is supplied from a municipal central heating plant at 165 lb pressure, reduced to 5 lb in a single step, which is the pressure used on all heating equipment throughout the building, except direct radiation which is supplied 1 lb steam.

Types of Motors Available

The range of seasons in St. Louis making it necessary to have the air conditioning plant as flexible as possible and the desire to keep the operating cost low whenever possible influenced the selection of motors. Four types of alternating current electrical motors are available from which to choose to meet the requirements, three of these being of the squirrel cage type and the other the slip ring motor. The squirrel cage motors are known as normal torque, normal starting current; normal torque, low starting current, and high torque, low starting current induction motors.

The normal torque, normal starting current motor is one that is designed with the aim that no one characteristic be over developed at the expense of the others. It develops full starting torque when connected directly across the line on full voltage but the starting current is not necessarily within the limits specified by the *American Institute of Electrical Engineers* and the *National Electrical Manufacturers Association*. Whenever central stations require current limiting starting equipment on such motors the starting torque is less. These motors, started with current limiting equipment, are suitable for applications where part of the load is applied after the motor is running at full speed, such as compressors, centrifugal pumps, blowers, machine tools, etc., where the load increases with the speed and the full load is applied after the motor is started.

The **high torque, low starting current motor** is capable of developing a high starting torque approximately 25 per cent greater than the normal torque motor on full voltage with the starting current approximately 10 per cent less than the normal torque motor started on full voltage but within the required limits on 30 hp sizes and smaller. These motors are started directly across the line through magnetic starters or other approved convenient starting devices.

The normal torque, low starting current motor has the same torque as the normal torque, normal starting current motor on full voltage but with the starting current approximately 20 per cent less than the normal torque motor on full voltage and well within the required limits on 30 hp sizes and smaller.

These three motors are wound for these different specifications to meet different load requirements, and are available for multi-speed operation, which must be considered for the maximum flexibility.

In addition to these three squirrel cage motors, the so-called **slip ring or wound rotor motor** is capable of very high torque with slower acceleration and the starting equipment may be arranged for starting only or for adjustable variable speed operation and slower even acceleration.

This, in general, covers the types available and it is important that each piece of driven equipment have the right motor in order that the whole installation may operate with the greatest efficiency and continuity of service.

Selecting the Proper Motor

The motors for the heating, ventilating and air conditioning equipment were selected to keep the maximum electric current demand as low as possible and the advantage of every low starting load requirement was taken to this end. Also, multi-speed motors were used freely to add flexibility to the operation and the ability of the installation to meet varying conditions and a wide season change. The twenty fan motors located in the ten fan rooms for both the exhaust and supply fans are two-speed, with the exception of two three-speed motors on supply fans (units 3 and 4) which gives greater flexibility in air supply to the basement exposition hall.

These multi-speed motors are connected through magnetic starting equipment with overload and low voltage protection, directly across the line, with com-

pelling relays to insure starting on low speed regardless of the ultimate speed setting at the main switchboard, where speed adjustment and stopping are controlled. This again keeps the maximum demand low without interfering with flexibility or operation. Since the motors are started at the motor, manual control could have been used but not at a lower cost for the same protection.

All multi-speed motors whose slow speed size is 7½ hp or larger, are normal torque, low current motors. Therefore, current limiting starting equipment is unnecessary because the current is within the limits for full voltage across-the-line starting and the low horsepower draws relatively low current from the line.

The pump motors for the dehumidifying apparatus (one in each fan room) are single speed, normal torque, low current motors. These motors drive centrifugal pumps which again make it possible to use the low current motor and lower the maximum demand load.

It would have been possible to select a slip ring type variable speed motor for driving fans instead of the multi-speed squirrel cage type. The slip ring motor, however, has much lower efficiency when operated at slow speeds than the multi-speed squirrel cage type and the control equipment is considerably more complicated. It would be possible, of course, with the slip ring motor to obtain a larger number of speeds, but since two or three speeds are all that are necessary to accomplish the desired purpose, it is apparent that the multi-speed squirrel cage type is better suited to the particular application.

The horsepower ratings of the fan motors on the lowest speeds are one-fourth of the horsepower rating at the highest speeds. For example, a fan motor rated at 40 horsepower at 1200 rpm is designed as a 10 horsepower motor for the 600 rpm speed. Magnetic starters are provided for these motors and compelling relays are installed on the starters to compel starting of the motors on low speed. In this way advantage is taken of the low horsepower rating of the low speed winding to reduce the starting current required.

Incidentally, in order to insure good voltage regulation on the motor circuits in the building, without the necessity of using exceptionally large section copper wire from the main distribution point in the building to the motors, all motors 1 hp and larger are served at 440 volts.

As mentioned previously, all pump motors and exhaust and supply fan motors are controlled by magnetic starters. Any motor can be started or stopped from a push button station located adjacent to the motor itself. In addition, any of the multi-speed motors can be changed in speed or stopped and any pump motor can be stopped from a push button control panel located in the main machinery room. Note, however, that it is not possible to start any motor from the main machinery room. It is necessary for an operator actually to be in the fan room in order to start any fan or pump motor, which insures an opportunity to the attendant of observing whether all equipment is functioning properly.

The motors operating the air filters are interlocked

(*Continued on page 77, Back Section*)

MOTORS INSTALLED IN THE ST. LOUIS AUDITORIUM

Motors in Main Machine Room

Two 3 hp, 1750 rpm, normal torque normal current motors driving circulating pumps—one for hot and one for cold water.

Two 10 hp, 1750 rpm, normal torque low current motors driving house pumps for pressure tanks on the roof.

Six 10 hp, 1750 rpm, normal torque low current motors driving condensate pumps.

Two 15 hp, 3600 rpm, normal torque low current motors driving vacuum cleaners.

One 60 hp, 1750 rpm, normal torque normal current motor driving air compressor air power for exposition hall.

One 100 hp, 1750 rpm, slip ring motor driving fire pump.

Two 3 hp, 1750 rpm, normal torque normal current motors driving compressors for damper control.

Motors in the Ten Fan Rooms Driving Dehumidifier Pumps

Two 25 hp, 1750 rpm, normal torque low current motors. (Pump capacities are 1122 & 1072 gpm.)

Three 20 hp, 1750 rpm, normal torque low current motors. (Pump capacities are 795, 795 & 900 gpm.)

One 15 hp, 1750 rpm, normal torque low current motors. (Pump capacity is 728 gpm.)

Two 10 hp, 1750 rpm, normal torque low current motors. (Pump capacities are each 524 gpm.)

Two 3 hp, 1750 rpm, normal torque normal current motors. (Pump capacities are 143 & 120 gpm.)

Motors in the Ten Fan Rooms Driving Fans

Three 40/10 hp, 880/430 rpm, normal torque low current motors, one in each of the three fan rooms. (Fan capacities are 75,000, 68,000 & 68,000 cfm.)

Three 15/3.7 hp, 870/430 rpm, normal torque normal current motors driving 56,500-cfm exhaust fans—one in each of three fan rooms.

Two 30/17/4.2 hp, 1160/880/440 rpm, normal torque low current motors driving 60,000-cfm supply fans—one in each of two fan rooms.

Four 10/2.5 hp, 870/440 rpm, normal torque normal current motors driving 42,000-cfm exhaust fans—one in each of four fan rooms.

One 20/5 hp, 1150/570 rpm, normal torque normal current motor driving 75,000-cfm exhaust fan.

One 30/7.5 hp, 1160/580 rpm, normal torque low current motor driving 60,000-cfm supply fan.

Two 7½/1.9 hp, 1150/570 rpm, normal torque normal current motor driving 11,000-cfm supply fans.

Two 5/1.3 hp, 1150/570 rpm, normal torque normal current motors driving 16,600-cfm exhaust fans.

Two 25/6.2 hp, 1160/580 rpm, normal torque low current motors driving 78,800-cfm supply fans.

Motors in the Ten Fan Rooms Driving Self-Cleaning Filters

Ten ¼ hp, 1200 rpm, single phase motors.

Motors Driving Exhaust Fans—for Corridors, Wash Rooms, Dressing Rooms, Etc.

One 7½ hp, 1160 rpm, normal torque low current motor. (Fan is 24,000 cfm.)

One 7½ hp, 860 rpm, normal torque low current motor. (Fan is 24,000 cfm.)

One ½ hp, 1165 rpm, single phase motor. (Fan is 1250 cfm.)

Two ¼ hp, 1165 rpm, single phase motors. (Fans are 1000 cfm.)

One 1½ hp, 1750 rpm, normal torque normal current motor. (Fan is 5000 cfm.)

One 2 hp, 1750 rpm, normal torque normal current motor. (Fan is 6000 cfm.)

Two 1 hp, 1750 rpm, normal torque normal current motors. (Fans are 3500 cfm.)

Two ½ hp, 1200 rpm, single phase motors. (Fans are 2000 cfm.)

Motors on and near the Stage Driving Stage Equipment

One 25 hp, hoist motor—raises and lowers the orchestra pit.

One 7½ hp, hoist motor—raises and lowers 32-ton fire- and sound-proof curtain for arena side of stage.

One 5 hp, hoist motor—raises and lowers 25-ton fire- and sound-proof curtain for theater side of stage.

One 7½ hp, 1500 rpm, hoist motor—arena act curtain.

One 5 hp, 1500 rpm, hoist motor—theater act curtain.

One 5 hp, 1500 rpm, hoist motor—light bridge hoist.

Two 1½ hp, 1750 rpm, gear motors—draw curtains, one for arena and one for theater.

One 1½ hp, 1750 rpm, single phase gear motor on a portable winch used in raising scenery.

Two motor generator sets with a double-ended 3 kw, 125 volt, generator on each end for stage lighting control.

Note: This list is not entirely complete as more equipment is being added.

"Air conditioned babies," born in the air conditioned obstetrical room of the Crawford W. Long Memorial Hospital, Atlanta, Ga.

Photo, Frigidaire Corporation

Hospital Association Committee Reports on

Air Conditioning Existing Hospitals

IF THE hospital, true to its traditions, is to provide for patients all of the resources which science has found to be beneficial, it is obviously faced with the necessity of "taking some steps to improve its air," states the *American Hospital Association's* Committee on Air Conditioning in the introductory paragraphs of its recently published report.[1] However because probably no hospital can afford a complete air conditioning system for its entire building—the present cost of maintaining and installing air conditioning equipment being "excessive"—and because medical investigation of controlled air therapy is hardly past its preliminary stages, the report treats the subject from a popular standpoint, considering primarily what can be done to raise comfort levels, incidentally achieving more healthful conditions for the patient and those who care for him. When medical research has determined definitely the therapeutic values of air conditioning, and the equipment has been simplified and rationalized in cost, the hospital will be in better position to consider the wisdom of an investment in general air conditioning, states the committee.

Among the advantages of air conditioning to a hospital mentioned in the discussion is relief of hay fever and asthma of certain types when sufferers can be kept in pollen free rooms with properly tempered air; keeping up the physical efficiency and nervous energy of surgeons and nurses, and preserving the patient's vitality, with operating room temperatures and humidities under positive control and fumes and odors promptly removed; elimination of danger from gas explosions caused by static electricity when relative humidity is maintained at 55 per cent; saving in summer heat many a patient—post-operative, pneumonia, cardiac—hovering between life and death by the added stimulation which results from dropping the room temperature a few degrees and reducing the humidity. Ionization equipment and portable electric air purifiers are pointed out as particularly desirable in the autopsy room, and in rooms of patients suffering from maladies which produce so much odor that ordinary ventilation is not sufficient.

The committee states that air conditioning may be defined as the control of indoor climate by mechanical processes (perhaps some day chemically), and discusses briefly moisture control, dewpoint temperature, air motion, and air cleanliness. The report then describes types of equipment—which are classified as central, built-in room unit, and portable unit. It is stressed that the hospital should consider air conditioning in its entirety with an eye to the future, and that the first step is a thorough survey by a practical "air-minded" consulting engineer.

In considering humidity control, the committee points out that the average hospital can materially improve atmospheric conditions in winter by the installation of humidifying equipment, recommends a central spray unit with duct distribution of the air; for cooling and dehumidification in summer, the needs of the average hospital can be met most economically at the present time by a combination of built-in and portable units, the committee believes.

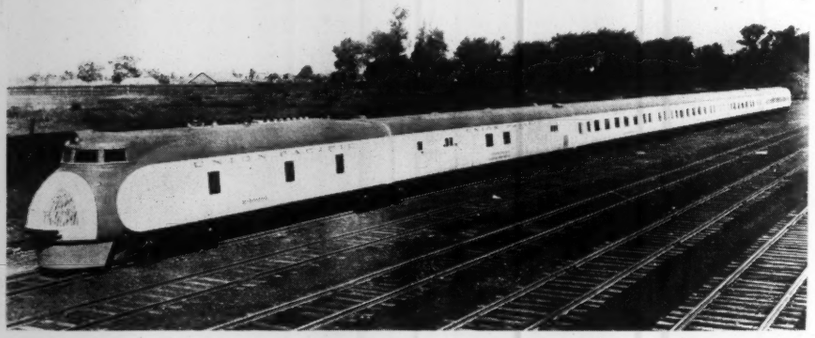
The report concludes with an estimate of the cost of providing a *reasonable minimum* of air conditioning for an existing building of 100-bed capacity the total capital cost approximating $6,400. No attempt is made to guess at the annual operating cost; however the committee stresses a point here that should be considered in not only hospitals but many other types of buildings—that in many instances the annual expense of partial air conditioning can be met with little if any increase in the engineering budget by studying and controlling the utilization of fuel, reducing waste caused by over-heating, stopping steam line, valve, and trap leaks, and insulating walls and roofs.

The committee defers offering suggestions for air conditioning new hospitals, confines its report to the mechanical and financial aspects of air conditioning existing buildings.

Members of the committee are Dr. C. W. Munger, chairman; Charles F. Neergaard; Dr. Lucius R. Wilson; and Perry W. Swerm.

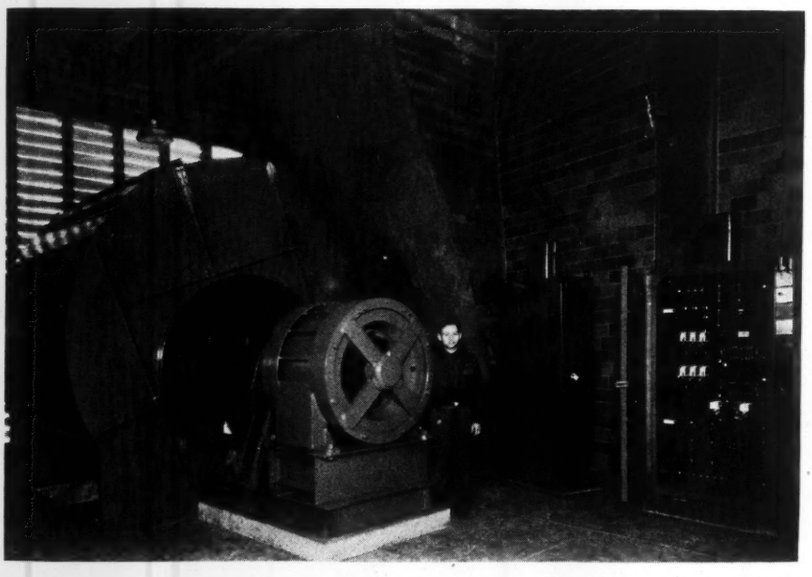
[1] "Report of the Committee on Air Conditioning" (Bulletin No. 119), published by the *American Hospital Association, Inc.*, 18 East Division St., Chicago. 9 pp., 6x9. Price, 50c.

IN THE NEWS . . . .



THE UNION PACIFIC STREAMLINER which streaked across the continent from Los Angeles to New York in less than 57 hours is, being modern, air conditioned. Fore-runner of the six-car record-breaker was the three-car stream-lined train described in the March, 1934, H. P. & A. C. Two nine-car stream-liners are now under construction, will go into service next year with the six-car speedster

SHOW ROOMS OF THE ELECTRIC HOME and Farm Authority in the James Building, Chattanooga, Tenn., offer a working demonstration of air conditioning. The illustration above shows the grilles just below the ceiling line (see arrow). Note how they were blended with the decorative lettering to fit in with the architectural scheme



A MILLION CUBIC FEET OF AIR PER minute is the capacity of the ventilating system for the East Boston Traffic Tunnel, said to be the largest vehicular tube in the world. The picture at the right shows one of the 28 fans, which are driven by two-speed squirrel cage motors direct connected by flexible couplings

Photo, The Lincoln Electric Co.

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

1895

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Tests of Three Heating Systems in an Industrial Type of Building

By G. L. Larson[1], D. W. Nelson[2], *and* John James[3], Madison, Wis.
(MEMBERS)

This paper is the result of research conducted by the Mechanical Engineering Department at the University of Wisconsin in co-operation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' *Research Laboratory.*

THE manner in which heat is distributed in a heated space is known to have considerable influence on the cost of heating. Maintenance of a uniform desired temperature in the occupied zone and a minimum temperature above that zone is conducive to efficient heating. The present program was undertaken to determine the effectiveness of heating with three different systems. The tests were conducted in a 71 by 182 ft one-story saw-tooth structure attached to the Mechanical Engineering Building and which is occupied by the Heating and Ventilating Laboratory. The side walls are of brick and one end wall is of brick and stone. It is considered to represent the usual industrial building of this type.

Description of Building

An aerial view of the Mechanical Engineering Building is shown in Fig. 1. The main structure is three stories in height and surrounds the Heating and Ventilating Laboratory on the west, north, and east. The influence of winds from these directions is, therefore, problematical. These surrounding walls also exert considerable influence on the sun radiation received. A plan and elevation of the Laboratory is shown in Fig. 2. It is attached to the main building at the north end and also at the southeast through the boiler room. There is a heated basement under one-half of section No. 3 at the south end. The remainder of the concrete floor rests on the ground.

Fig. 3 shows an interior view of the Laboratory. It is one large open space except for a partition located one-third of the distance from the north end and which extends one-half way across the width of the room. A 7 by 10 ft opening was made through this partition to allow free circulation of air.

Table 1 is a summary of the various heat loss items. The heat losses have been calculated using the temperature gradient of 2 per cent per foot for direct radiation and 1 per cent per foot for unit heater systems as recommended in THE A. S. H. V. E. GUIDE, 1934. This table shows that the roof and skylight heat losses are considerably over half of the total.

Description of Heating Systems

The Laboratory Building as originally built, which was previous to the construction of the remainder of the Mechanical Engineering Building, was heated by a two-pipe vacuum system using direct radiation. This consisted of wall radiation placed along the walls and in the trusses directly below the skylights. The location of these radiators is shown in Fig. 2. The total direct radiation surface in this system was 4,431 sq ft at the time of test. Fifty-three per cent of this amount is located directly below the skylights.

For the purpose of these tests, seven suspended unit heaters were installed and were located 10 ft from the floor in the arrangement shown on Fig. 2. These heaters were loaned for the purpose of test by the manufacturers. The total capacity as selected from the catalogs was 4,676 sq ft with 218 F steam and 60 F entering air; the heaters were selected to have approximately equal capacities.

Each of these seven units was tested for steam and air capacity in a test apparatus. These rating tests were conducted during the year 1932-1933 and were made in accordance with the Standard Code for Testing and Rating Steam Unit Heaters adopted by the Society in January, 1930. The test set-up is shown in Fig. 4. The results of the rating tests are given in Table 2. The total heat capacity was found to be 1,036,700 Btu or 4,318 sq ft equivalent direct radiation with 218 F steam and 60 F entering air. The total air moving capacity was 17,027 cfm. The total manufacturer's rating was 1,121,905 Btu and 20,265 cfm.[4]

Two floor unit heaters of identical size and construction were installed as a third heating system. Fig. 2 shows the location of these units and one of them appears in the photograph in Fig. 3. The total manufacturer's rating was 910,000 Btu or 3,790 sq ft E.D.R. at 218 F steam and 60 F entering air.

The same mains and branches were filled with steam in each of the tests of the three heating systems. The piping was in excess for the unit heater systems, since the branches to the various radiators were included. The

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[2]Assistant Professor of Steam and Gas Engineering, University of Wisconsin.

[3]Research Fellow in Heating and Ventilating, University of Wisconsin.

[4]Acknowledgment is due to C. H. Novotny, as a graduate student, for his cooperation on the rating tests.

For presentation at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935.

branches added to the original system for the unit heaters were comparatively short.

The Thermostatic Control System

A thermostatic control system, electrically operated, was installed so as to control the temperatures during all tests on each of the three heating systems. This thermostatic control was loaned for the purposes of this test by the manufacturers. Double or two-temperature thermostats were installed on three posts on the center line of the building so as to divide the space into three equal zones. A system of switches was installed so that the control could be shifted easily from one heating system to another. Fig. 2 shows the location of the thermostats and the units controlled by each. An electric clock unit shifted the circuits between the night and day thermostats when operating on dual or two-temperature control.

The thermostats opened and closed steam valves on the direct radiation, started and stopped the fan motors in the case of the suspended unit heaters only or in conjunction with the opening and closing of the steam valves, and for the floor unit heaters shifted the position of the by-pass damper. The fan motors on these floor units ran continuously. Since there were only two floor units placed at opposite ends of the Laboratory, only the two end thermostats were used. On certain tests of these floor units, thermostats were placed directly in front of the units and in the path of the return air.

Method of Testing

Steam consumption readings were taken daily at 5 p.m. on a calibrated condensation meter. Recording thermometers were installed in each of the three sections so that the sensitive elements were adjacent to heating units or in the path of the heated air from the units under test. By this means the *off* and *on* periods of heating were recorded.

Recorders were attached to the by-pass damper shafts of the floor unit heaters to show their position at all times. A recording thermometer was used to show the variations in temperature at the 5 ft level. This was moved about but usually was near the thermostat station in section No. 1. Fig. 5 shows typical charts of the temperatures at the 5 ft level for the three systems under dual control. Resistance thermometers were placed vertically at 4 ft intervals from floor to ceiling at a central location in each of the three zones as indicated on the floor plans of Fig. 2. The averages of the readings from floor to ceiling are shown for the three systems and the three sections on Fig. 6. As indicated on this figure each point represents a large number of readings taken on a great many days. Table 3 is a tabulation of all these readings to indicate the spread of temperatures from floor to ceiling in each section and for each system. The averages used in constructing curves of Fig. 6 are shown at the bottom of this table.

It was necessary to correct the results to those obtainable on a standard day in order to make comparisons between the quantities of steam used with the various systems. The standard day was assumed as being one with a 39 F outside temperature. This was chosen as being the mean outside temperature for Madison during an average heating season. The corrections were made in direct proportion to temperature differences. The steam consumption was multiplied by the ratio of difference between the inside temperature on a standard 39 F day and the outside temperature on the

Fig. 1—Aerial view of Mechanical Engineering Building at the University of Wisconsin

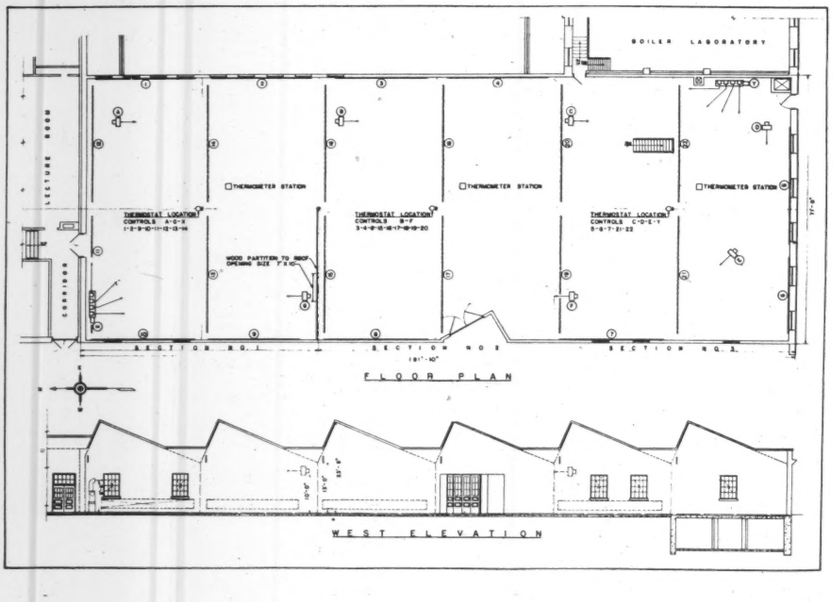

Fig. 2 (Right)—Floor plan and elevation of the Heating and Ventilating Laboratory

Table 1—Heat Losses and Surfaces for the Heating and Ventilating Laboratory

SURFACE		AREA	HEAT LOSS COEFF.	DIRECT RADIATION		UNIT HEATER	
				TEMP. DIFF.	HEAT LOSS B.T.U. HR.	TEMP. DIFF.	HEAT LOSS B.T.U. HR.
WALL	EAST & W.	5449 SQ.FT.	0.385	86.0 F.	180,500	83.0	174,300
	SOUTH	726 "	0.259	82.1	15,460	81.1	15,260
GLASS	LOWER	639 "	1.13	82.7	59,750	81.4	58,800
	ROOF	2898 "	1.13	101.7	333,000	90.8	297,500
FLOOR	NORTH	10254 "	0.1473	15.0	22,650	15.0	22,650
	SOUTH	2050 "	0.452	20.0	18,550	20.0	18,550
ROOF		12796 "	0.386	98.9	494,000	89.5	442,000
LEAKAGE	WALL	6175 "	0.0756	86.0	40,200	83.0	38,770
	GLASS	190 FT.	1.94	82.7	30,500	81.4	30,000
	DOOR	73 FT.	3.80	80.0	22,200	80.0	22,200
TOTAL					1,216,800		1,120,030

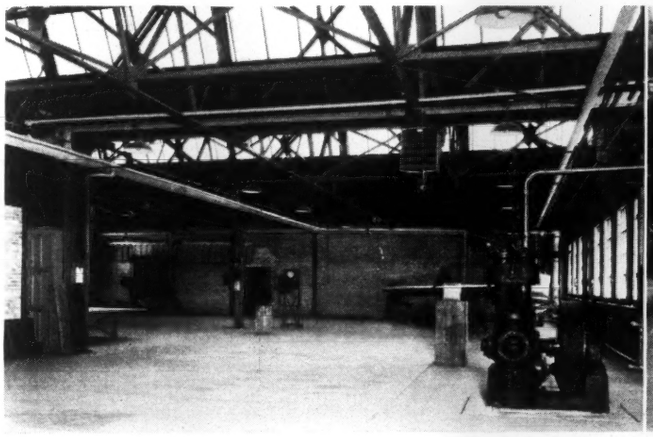

Fig. 3—Interior view of the Heating and Ventilating Laboratory

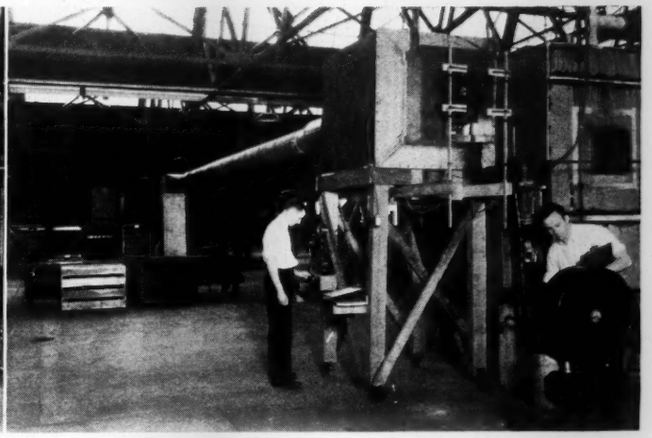

Fig. 4—General view of unit heater test code apparatus

standard day over the inside temperature minus the outside temperature on the day of test. The inside temperatures for this purpose were taken at the 5 ft level. In securing the probable inside temperature that would exist on a standard 39 F day, the inside temperatures for all tests were plotted against the outside temperatures. From this it was determined that a change in outside temperature did not affect the inside temperature at the 5 ft level when continuous control was used. With dual control, it was found that lowering the outside temperature 10 deg lowered the inside temperature 1.9 F. This correction was accordingly made on dual tests to obtain the probable inside temperature on a standard day.

The weather conditions were obtained from reports of the local Weather Bureau station located on the top of one of the campus buildings. Table 4 shows various factors in the weather for Madison over a long time period and for the season 1933-1934 in which these tests were made. This is included to make it possible to compare weather conditions of the tests with those in any other part of the country. In this table as well as in tables and figures showing results of tests, the sun radiation intensity and wind velocity are included, since they exert considerable influence.

Record of Test Results

The results of the tests are shown graphically in Figs. 7 to 12, inclusive. In all of these the daily steam consumptions are plotted against the daily average outside temperatures. Each point shown is accordingly the result of a 24 hour test. The average results for each test period of from 5 to 9 days is shown as a cross with the identifying test number in a circle. These averages were corrected to a 39 F standard day by the previously mentioned method and indicated by circles at 39 F. The curves were plotted through this standard day point and with due attention to the averages of the tests concerned.

In general, the location of a daily point above the curve indicates a higher than average wind or a lower than average sun intensity, or both, and a location below, the reverse. Unfortunately, the wind velocities and sun intensities for individual days cannot be shown within the limits of this paper. The averages for test

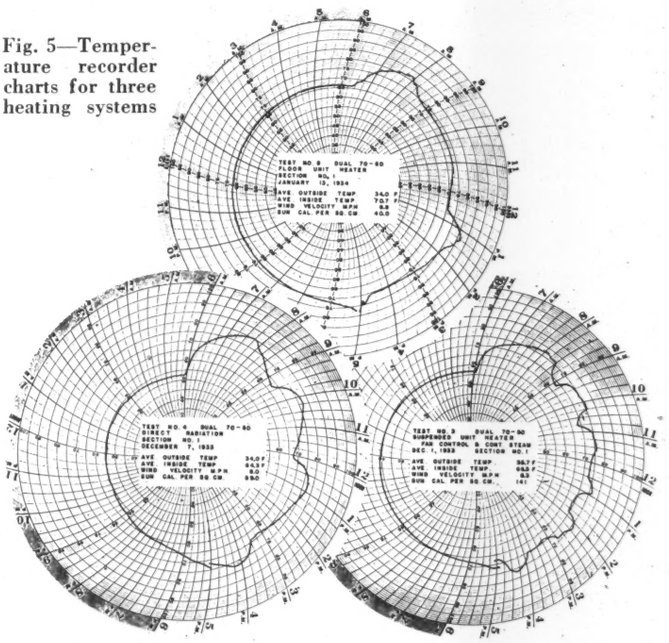

Fig. 5—Temperature recorder charts for three heating systems

Table 2—Comparison of A. S. H. V. E. Code Test and Manufacturers Catalog Ratings for Suspended Type Unit Heaters

HEATER NO.	SPEED	B. T. U. % DIFF.	FINAL AIR TEMP. ACTUAL DIFF.	C. F. M. % DIFF.	WATTS PER 1000 C.F.M.
1	HIGH	+4.9	+9.7 F	+ 3.6	105.2
1	LOW	+4.4	+4.5 F	+ 3.3	
2	HIGH	+0.8	−6.2 F	+16.2	180.0
2	LOW	−4.5	−11.3 F	+14.7	
3	HIGH	+8.4	−8.7 F	+28.5	113.4
3	LOW	+18.3	−11.5 F	+38.2	
4	HIGH	−9.0	−10.9 F	+15.7	158.3
4	LOW	−7.2	−12.3 F	+17.0	
5	HIGH	+2.1	−4.5 F	+19.4	167.2
5	LOW	−5.6	−9.5 F	+16.4	
6	HIGH	+52.3	+5.4 F	+28.8	155.4
6	LOW	+38.9	−3.6 F	+60.9	
7	HIGH	+5.1	+0.4 F	+17.4	140.3
7	LOW	−2.8	−3.4 F	+12.3	

NOTE:- PERCENTAGES ARE ON BASE OF TEST VALUES.
+ SHOWS CATALOGUE RATING HIGHER THAN TEST RATING.

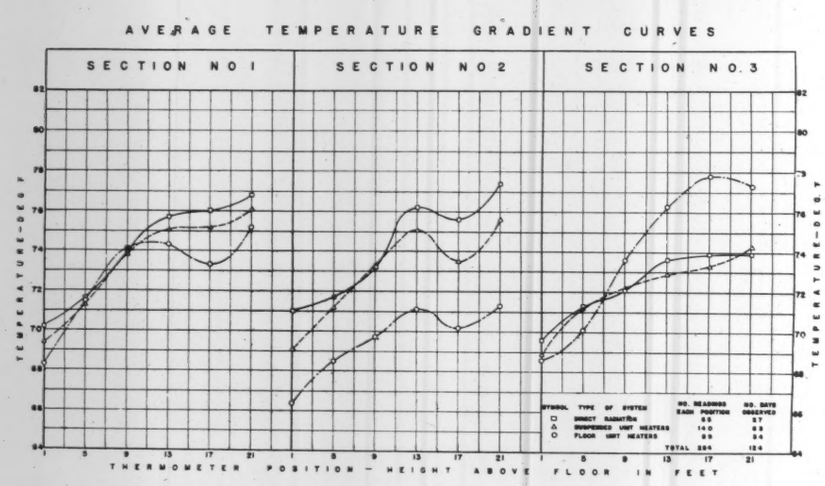

Fig. 6—Temperature gradient curves for three heating systems

periods, however, are shown on these figures and in general wind and sun intensity differences explain the location above or below the curve.

Table 5 is a tabulation of the results of the three heating systems with continuous and dual control. The steam consumptions corrected to conditions of a standard day of 39 F are shown here for all tests. The differences in steam consumption are given in percentage values.

Dual and Continuous Operation of Systems

The direct radiation system used 10,205 lb per day corrected to a standard 39 F day. When operated on dual control with the day thermostat set for 70 F for 11 hours and the night for 50 F for 13 hours, the consumption was 8,700 lb per day. This is a saving for dual control of slightly less than 15 per cent. The wind velocity was 11.4 mph and the sun intensity was 380 cal per sq cm for the continuous tests. For the corresponding dual tests the wind velocity was 9.8 mph and the sun intensity was 121 cal per sq cm. These are in the direction to offset each other. It is considered that the sun intensity difference of 259 has a greater influence than the wind velocity difference of 1.6. The difference between continuous and dual control is therefore probably somewhat greater than the 14.7 per cent shown in the table.

The tests on the suspended unit heater system with fan and steam control indicate a saving of almost 31 per cent for dual over continuous operation. The higher wind velocity and lower sun intensity during the dual tests over those of the continuous tests indicate that had these factors been equal in the two tests the saving would have been higher than 31 per cent.

With thermostatic control of the fan motor only, the steam being on the units continuously, a saving of 27.5 per cent is indicated for dual over continuous control. The wind velocities were equal. The sun intensity was slightly higher during the dual tests, which would indicate that the steam consumption should have been somewhat higher on the dual tests and the savings therefore somewhat less than the 27.5 per cent shown in the table. As these results indicate, it would seem that the saving for dual control should be less when the steam is maintained on continuously day and night than when the dual control is applied to the control of steam as well as to the fan operation.

As mentioned previously, the 14.7 per cent saving for dual operation of direct radiation is considered low because of the difference in sun intensities of 380 and 125 cal per sq cm. Had these sun intensities been equal, the saving might have been nearer those shown for the suspended unit heater systems. This saving for dual control with direct radiation also may have been reduced compared to those for unit heaters by the fact that lower outside temperatures existed during these tests over those for the unit heater tests. This would cause steam to be on the radiators longer during the night period to maintain the set temperature, with consequent over runs in temperatures. This would seem to cause the steam consumption on a standard day to be higher than it would have been had the outside temperature at the time of test been nearer to the standard day temperature of 39 F.

The saving of steam due to dual control when maintaining 60 F at night with the floor unit heaters was 16 per cent. The wind velocity was slightly lower on the dual test than on the continuous test. The sun intensity was considerably less, which probably more than offset the difference of wind velocities. It is likely, then,

Table 3—Temperature Gradient Readings for the Three Heating Systems

TEMP.	DIRECT RADIATION			SUSPENDED UNIT HEATERS			FLOOR UNIT HEATERS		
	SECTION NO. 1	SECTION NO. 2	SECTION NO. 3	SECTION NO. 1	SECTION NO. 2	SECTION NO. 3	SECTION NO. 1	SECTION NO. 2	SECTION NO. 3
	THERMOMETER HEIGHT ABOVE FLOOR - FT.								
	1 5 9 13 17 21	1 5 9 13 17 21	1 5 9 13 17 21	1 5 9 13 17 21	1 5 9 13 17 21	1 5 9 13 17 21	1 5 9 13 17 21	1 5 9 13 17 21	1 5 9 13 17 21
62–			1				1		
62								2 2 2 2 2 1	1 1 1 1
63	1							8	1 1 1
64			1					1 7	1 1
65			4 3 2 2 3 4	3				2 7 3	3 1
66	2 2	2 2 5 2 1 3	2 2 5 2 1 3	5 3	8			8	
67	2 1 1 1 1 4 2 1		4 3 2 2 3 4	15 3	15			15 4 4 7 1 6 2	
68	6 2 1 1 6 2 1 2	3 2 2 5 3 2	3 2 2 5 3 2	21 8 2 1	33 3		7 1	29 10 3 3 6 3	5 2 1 1 1 15 8 6 3 4 5 3 2
69	8 7 1 1 12 4	2 12 1 1 2 5 2	3 1 2 5 2 3	1 3 10 7 1 1	16 6 13 9 8 4	9 8 8 10 10 7	12 15 13 7 7 7 9 12 7 4 4 5		
70	16 11 8 3 5 2	15 10 7 3 9 1	19 13 6 5 3 6	32 22 11 5 4 2	31 10 9 3 1	35 25 16 14 19 17	6 16 11 5 7 5 4 14 13 11 7 12 15 8 9 8 7		
71	10 9 8 6 5 6	11 11 10 6 4 9	8 10 7 7 7 5	17 26 11 3 5 2	15 35 16 2 11 5	16 28 19 17 12 13	10 6 7 10 6 11 1 7 16 10 12 9 10 10 7 5 7 6		
72	10 7 6 7 4 3	7 11 6 6 4 3	4 10 7 4 3 3	10 22 26 17 17 6	7 28 22 11 14 6	9 26 25 24 23 15	1 11 13 12 9 7 2 6 6 14 14 6 8 11 13 9 6 6		
73	6 12 10 6 7 5	5 10 8 4 4 4	6 10 9 5 6 4	5 23 19 19 18 17	1 14 26 18 35 11	3 16 21 21 13 14	6 13 6 5 9 11 1 9 8 8 20 3 7 3 3 6 7		
74	1 8 7 6 3 5	1 6 9 6 4 4	1 5 10 5 4 4	13 3 23 17 21	3 17 19 28 23	10 17 18 18 15	1 6 5 4 3 3 3 4 1 1 5 6 3 5 3 11 8 5 5		
75	2 3 3 7 5 4	5 6 3 3 4	3 3 8 9 9	4 9 11 17 12 1	1 16 38 21 22	5 13 14 11 11 13	5 4 6 3 8 2 2 1 3 5 4		
76	3 4 6 10 6	2 5 2 5 3	3 4 6 7 7	1 2 22 15 17 13	7 26 17 23 1	6 10 15 9	4 2 2 8 6 2 2 3 2 2 6 4 2 6		
77	1 7 3 2 5 1	4 7 9 2 1	3 5 3 4	1 16 18 22 16	4 12 9 19	2 2 4 5 6	4 7 5 3 4 2 1 2 7 5 2		
78	1 2 2 2 7	1 3 6 2 7	1 1 2 1	5 16 14 17	1 10 1 16	1 1 3 2 10	6 5 6 3 1 2 6 3 2 3		
79	3 4 5 3	1 2 6 4	1 1 3 1 4	4 5 6 13	5 1 12	1 2 3 7	1 4 4 6 7 2 2 2		
80	4 2 2 4	11 3 3	1 1 1	1 5 4 12	2	1 1 5	4 6 3 7 4 4 2 3		
81	4 3 2	1 2 5	2 1 2	2 4 5	1	1 2 4	3 3 2 1 6 4 2		
82	3 5 3	4 5 1	1 1 1	2			2 2 1 4 1 4 2 5		
83	3 3 2	3 2 1	1 1	1		1	2 4 10 4 7		
84	1 1 3	1 3					1 6 3		
85	1 1	5	1 1				4 5 6		
86	1	1	2 1			1	1 7 5		
87	-	2	1				1 2 2		
88							5 3		
88+						1 2	2		
AVE. TEMP.	732 716 739 757 760 768	710 717 732 762 756 774	694 713 721 736 739 739	694 713 738 751 752 761	691 712 723 751 735 734	689 712 722 729 723 743	683 715 741 743 733 732	644 685 687 711 702 713	684 701 726 763 778 773
	NO. READINGS AT EACH THERMOMETER HEIGHT = 65			NO. READINGS AT EACH THERMOMETER HEIGHT = 140			NO. READINGS AT EACH THERMOMETER HEIGHT = 89		

that the saving would have been greater than 16 per cent had the wind and sun factors been alike in the two tests.

With this same night temperature of 60 F, the suspended unit heaters showed 23.6 per cent saving. The sun intensity and wind velocity were both low in the dual test. However, the difference in sun intensities is greater, which indicates that the real saving would be larger.

Table 4—Monthly Weather Bureau Averages and Means for Madison, Wis.

MONTH	MEAN TEMPERATURE		WIND 1869 — 1930		WIND 1933 — 1934		SUN INTENSITY CAL. PER SQ. CM.	
	1869-1930	1933-34	AVE. VEL. M.P.H.	PREVAILING DIRECTION	AVE. VEL. M.P.H.	PREVAILING DIRECTION	NORMAL	1933-34
JANUARY	16.7	26.6	10.1	N.W.	9.8	N.W.	4,727	3,990
FEBRUARY	19.8	18.5	10.6	N.W.	9.7	N.W.	6,496	6,911
MARCH	31.0	29.9	11.3	N.W.	10.1	N.W.	9,820	9,627
APRIL	45.6	45.8	11.3	N.W.	11.1	N.W.	12,043	11,809
MAY-1934	57.6	65.7	9.9	S.	9.2	S.	14,706	16,934
JUNE-1933	67.1	76.9	8.2	S.W.	7.7	S.W.	15,553	17,333
JULY	72.2	73.6	7.7	S.W.	7.4	S.W.	16,243	16,009
AUGUST	69.9	69.6	7.6	S.	7.0	S.E.	13,892	15,291
SEPTEMBER	62.1	67.2	8.8	S.	8.8	S.W.	10,249	9,685
OCTOBER	49.9	49.0	9.9	S.	9.4	S.	7,102	7,725
NOVEMBER	35.0	34.0	11.0	N.W.	10.9	N.W.	4,323	4,326
DECEMBER	22.6	23.4	10.5	N.W.	9.4	N.W.	3,615	3,066

Fig. 7 shows graphically the data discussed on the direct radiation system with dual and continuous control. Fig. 8 shows the corresponding data for the suspended unit heater system.

Comparison of Direct Radiation and Unit Heaters

Suspended unit heaters with continuous thermostatic control showed a saving in steam consumption of 6.7 to 13.5 per cent over direct radiation. The 6.7 per cent was secured in the tests with control of the steam supply as well as of the fan motors. The 13.5 per cent was found with control of the fan motors only. The greater saving would be expected with control of both air movement and steam. The reason that this was not obtained is difficult to explain. The sun intensity and wind velocity effects tend to counteract each other, although probably the net result if they had been equal in the two sets of tests would have been an increase of the 6.7 per cent saving found with fan and steam control. The still air capacity of the unit heaters is very low, so that little difference should exist between the steam consumptions of the two methods of control.

On dual control tests, 24.3 and 26.5 per cent savings were found for the suspended unit heater system as compared to the direct radiation system. The greater saving of unit heaters on dual control over that for continuous control would seem to be due to the unit heater allowing faster cooling at the beginning of the night period, less overshooting of the night temperature when heat is required and lower temperatures near heat loss surfaces during the warming up period in the mornings. As mentioned under the dual and continuous comparison for direct radiation, it is considered that the steam consumption for continuous control with direct radiation was low because of abnormally high sun intensity. This seems to indicate that the saving of unit heaters over direct radiation in this building is about 25 per cent for dual control and something over 13.5 per cent, possibly 20 per cent for continuous control.

On the dual tests, as with the continuous tests, the complete control of air movement and steam resulted in less saving than the less complete control of air only. The difference, however, is very slight. The wind velocities were equal, but had the sun intensities been equal the percentage for the complete control would have been greater, as would be expected. Fig. 9 shows graphically the comparison between direct radiation and suspended unit heaters for continuous control and Fig. 10 shows the same for dual control.

The suspended unit heater system showed a saving of 16.5 per cent when a 60 F temperature was carried at night instead of 50 F. This is in comparison to direct radiation at 50 F. The floor unit heater system showed a saving of 10.2 per cent at 60 F night temperature, as compared to direct radiation at 50 F night temperature.

With continuous control the floor unit heater system showed a saving of 8.9 per cent over similar operation with direct radiation. The wind and sun effects balance

Table 5—Results for the Three Heating Systems

TEST NO	PERIOD OF TEST	NO. DAILY TESTS AVERAGED	SYSTEM	AVE INSIDE TEMP. F	AVE. OUTSIDE TEMP. F	AVE WIND VEL. M.P.H.	SUN INTENSITY CAL. PER SQ. CM.	AVE. TEMP DIFF IN - OUT	COR AVE IN. TEMP. MINUS 39 F STD. DAY	ACTUAL STEAM CONSUMPTION AVE LBS PER DAY	AVE. LBS. STEAM PER DEG. T. DIFF. IN - OUT	COR. STEAM CONSUMPTION AVE. LBS. PER STD. 39 F DAY	SAVING IN PER CENT OF CONTINUOUS CONTROL	SAVING OVER DIRECT RADIATION	SAVING OVER SUSP. UNIT HEATER
CONTROL — CONTINUOUS 70 F — 24 HOURS														PER CENT OF TESTS 14-17	PER CENT OF TESTS 6-13
14	MARCH 15-22	7	DIRECT RADIATION	72.3	31.9	11.0	345	40.4	33.3	13,130	326	10,830			
17	APRIL 7-14	7	DIRECT RADIATION	73.3	43.4	11.8	415	29.9	34.3	8,350	280	9,580			
			AVE.	72.8	37.7	11.4	380	35.1	33.8	10,740	306	10,205			
6	DEC. 21-29	8	SUSP. U.H. FAN & STEAM CONTROL	71.7	10.0	7.9	127	61.7	32.7	17,810	289	9,440			
13	MARCH 7-14	6	SUSP. U.H. FAN & STEAM CONTROL	73.8	24.6	10.2	337	49.2	34.8	13,650	277	9,640			
			AVE.	72.7	16.3	8.9	217	56.4	33.7	16,030	285	9,520		8.72	
12	FEB. 26-MAR. 7	8	SUSP. U.H. FAN CONTROL-CONT. ST.	73.2	30.7	10.6	249	42.5	34.2	11,600	273	9,330			
16	APRIL 1-7	7	SUSP. U.H. FAN CONTROL-CONT ST	74.0	42.2	9.5	234	31.8	35.0	7,490	236	8,250			
			AVE	73.5	36.1	10.1	242	37.4	34.5	9,680	258	8,830		13.47	7.25
9	JAN 18-27	7	FLOOR U.H. THERM AT UNIT	71.6	33.7	11.1	143	37.9	32.6	11,210	296	9,660			
10	FEB. 5-13	6	FLOOR U.H. THERM. AT UNIT	71.9	15.7	10.1	227	56.2	32.9	16,320	290	9,550			
11	FEB 13-22	8	FLOOR U.H. THERM. AT COLUMN	72.0	25.8	9.8	257	46.2	33.0	12,280	267	8,790			
15	MARCH 22-31	8	FLOOR U.H. DUAL THERM. AT COL.	73.5	28.8	9.2	304	46.7	34.5	12,430	266	9,180			
			AVE.	72.3	25.9	10.0	235	46.4	33.3	12,950	279	9,300		8.87	2.31
CONTROL — DUAL 70 F. 11 HR. DAY — 50 F. 13 HR. NIGHT														PERCENT OF TESTS 1-4	PERCENT OF TESTS 2-5
1	NOV. 11-18	5	DIRECT RADIATION	63.3	21.8	10.9	161	41.5	27.6	13,450	324	8,940			
4	DEC 3-12	8	DIRECT RADIATION	63.9	25.9	9.1	96	38.0	27.4	11,880	312	8,550			
			AVE.	63.7	24.3	9.8	121	39.4	27.5	12,510	317	8,700	14.73		
2	NOV. 18-25	7	SUSP. U.H. FAN & ST. CONTROL	65.4	37.6	9.9	139	27.8	26.7	6,575	236	6,300			
5	DEC 12-21	7	SUSP. U.H. FAN & ST. CONTROL	62.5	27.1	10.7	81	35.4	25.8	9,425	266	6,870			
			AVE.	64.0	32.3	10.3	110	31.7	26.2	8,000	252	6,590	30.78	24.25	
3	NOV. 25 - DEC. 3	7	SUSP. U.H. FAN CONTROL-CONT. ST.	65.5	37.3	9.8	128	28.2	26.8	7,750	275	7,375			
18	APRIL 15-25	9	SUSP. U.H. FAN CONTROL-CONT. ST.	68.9	42.5	10.2	388	26.4	29.2	5,105	193	5,640			
			AVE.	67.5	40.3	10.1	275	27.2	28.2	6,265	231	6,400	27.52	26.45	2.88
CONTROL — DUAL 70 F 11 HR. DAY - 60 F 13 HR. NIGHT															
7	JAN. 4-11	7	SUSP. U.H. FAN & ST. CONTROL	66.3	28.0	7.6	66	38.3	29.4	9,470	248	7,270	23.84	16.44	- 10.32
8	JAN. 11-18	7	FLOOR U.H. DUAL THERM. AT COL.	67.3	26.4	9.3	139	40.9	30.7	10,410	255	7,910	16.02	10.24	- 18.81

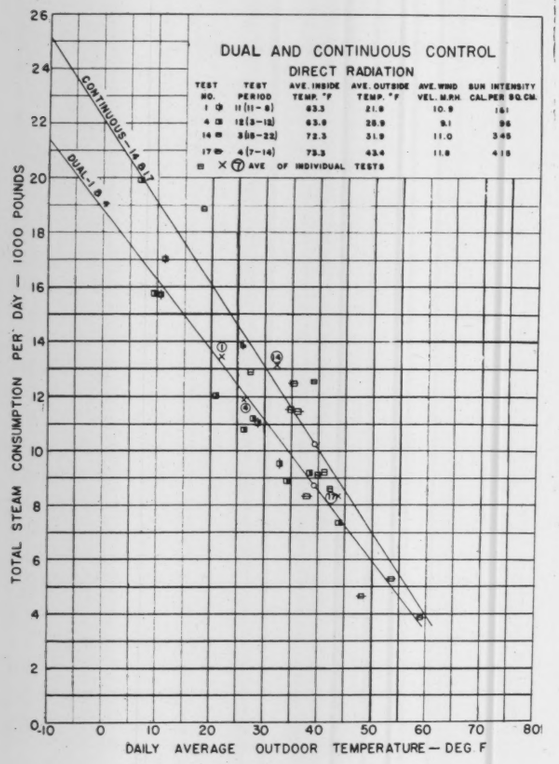
DUAL AND CONTINUOUS CONTROL
DIRECT RADIATION
TOTAL STEAM CONSUMPTION PER DAY — 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE — DEG. F

Fig. 7—Steam consumption curves for dual and continuous control with direct radiation

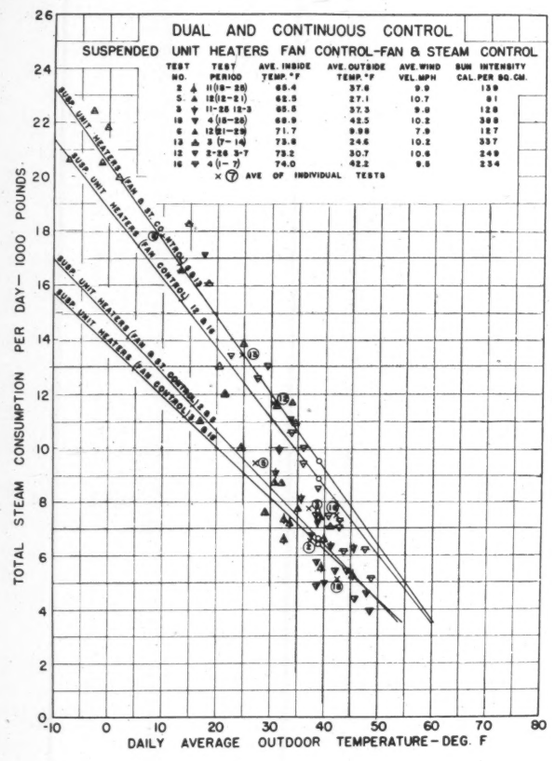
DUAL AND CONTINUOUS CONTROL
SUSPENDED UNIT HEATERS FAN CONTROL-FAN & STEAM CONTROL
TOTAL STEAM CONSUMPTION PER DAY— 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE-DEG. F

Fig. 8—Steam consumption curves for dual and continuous control with suspended unit heaters

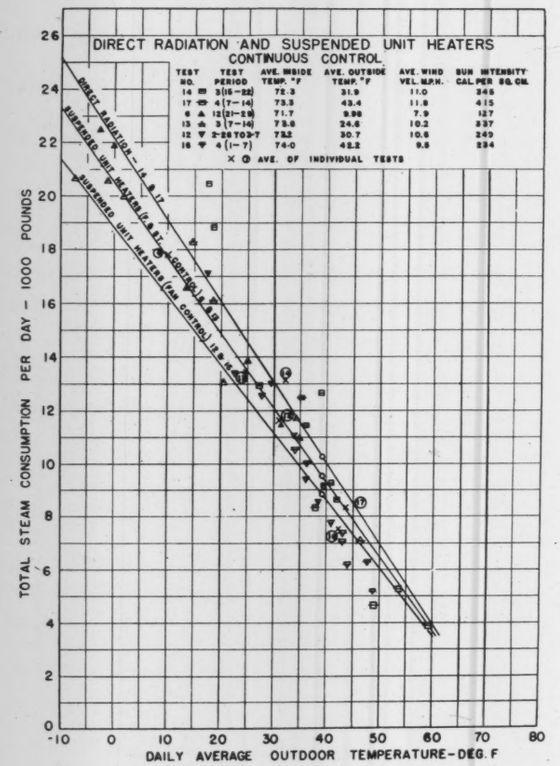
DIRECT RADIATION AND SUSPENDED UNIT HEATERS
CONTINUOUS CONTROL
TOTAL STEAM CONSUMPTION PER DAY - 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE-DEG. F

Fig. 9—Steam consumption curves for direct radiation and suspended unit heaters with continuous control

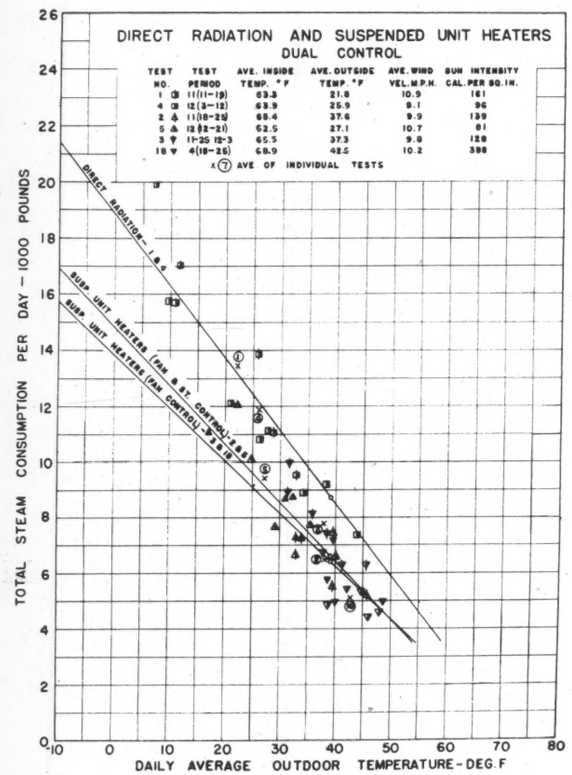
DIRECT RADIATION AND SUSPENDED UNIT HEATERS
DUAL CONTROL
TOTAL STEAM CONSUMPTION PER DAY — 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE-DEG. F

Fig. 10—Steam consumption curves for direct radiation and suspended unit heaters with dual control

each other to some extent, but the considerably greater sun intensity on the direct radiation test would make the real saving greater than the 8.9 per cent obtained. Fig. 11 shows these results graphically.

Comparison of Floor Unit Heaters and Suspended Unit Heaters

The steam consumption with the floor unit heater system was found to be practically the same as that for the suspended unit heater system. On continuous thermostatic control, the floor unit heaters showed a saving of 2.3 per cent in steam consumption over the suspended unit heaters. On these tests the sun and wind effects do not differ greatly and tend to offset each other. Fig. 12 shows the data graphically for this comparison. A comparison on dual control with a night temperature of 60 F showed 7.4 per cent more steam used by the floor unit heater system than by the suspended unit heater system. The sun and wind effects are quite different but are in the direction of offsetting each other. Should this difference of 7.4 per cent be real, it is considered to be due to leakage past the temperature control dampers of the floor unit heaters. The steam is maintained continuously on these units and the motors also operate continuously. On mild days a tendency to overheat the building was noted and this probably was occurring during the night periods on dual control.

Inside Temperatures and Temperature Gradients

The charts shown in Fig. 5 are daily records for the three heating systems for similar weather conditions. In each case the chart is for dual operation and was taken in section one at the 5 ft level within a few feet of the thermostat. The same thermostats and the same day settings were used except in certain tests of the floor unit heaters, where a thermostat was placed in the path of the return air. Night temperature settings of 50 F and 60 F were carried on tests involving dual contol.

For the days shown in Fig. 5, the night temperature did not fall below the thermostat setting. With direct radiation a minimum temperature of 52.5 F was reached and with the suspended unit heaters it was 56.5 F. The outside temperature during this unit heater test was somewhat warmer, but this higher inside temperature was due in part to the steam being in the units continuously. With the floor unit heaters the temperature dropped to 66 F. The steam was in the units continuously in these tests and the motors also ran continuously. This higher night temperature is considered to be due to leakage of air around the control dampers and conduction of heat from the steam coils to the air going through the by-pass passageway. The night temperature setting was 60 F in this case, but the record would appear the same had it been 50 F as in the other two charts, since the temperature did not fall as low as the night setting.

The rate of cooling on the direct radiation system was 6.3 F per hour over the first two hours of night setting, which starts at 5 p.m. For the suspended unit heaters with steam on continuously, it was 3.3 F per hour with a slightly higher outside temperature. With floor unit heaters, the cooling rate from 5 to 7 p.m. was 4 F per hour.

The direct radiation system required one-half hour

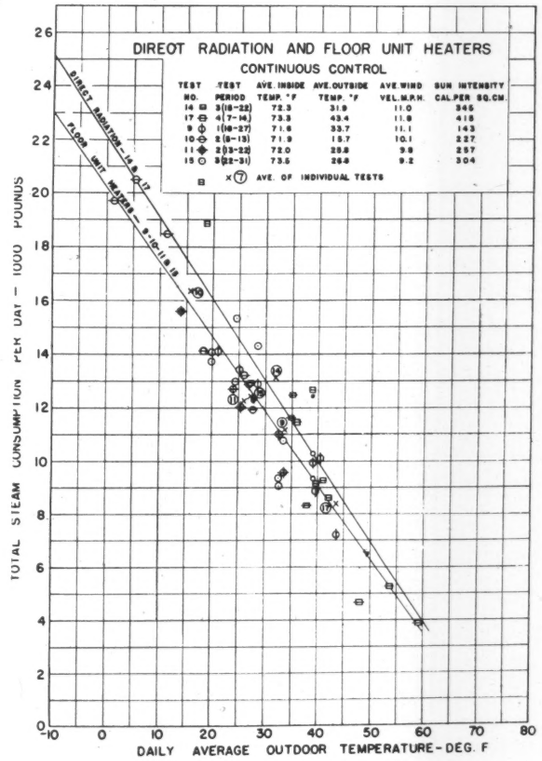
DIREOT RADIATION AND FLOOR UNIT HEATERS
CONTINUOUS CONTROL
TOTAL STEAM CONSUMPTION PER DAY - 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE - DEG. F

Fig. 11—Steam consumption curves for direct radiation and floor unit heaters with continuous control

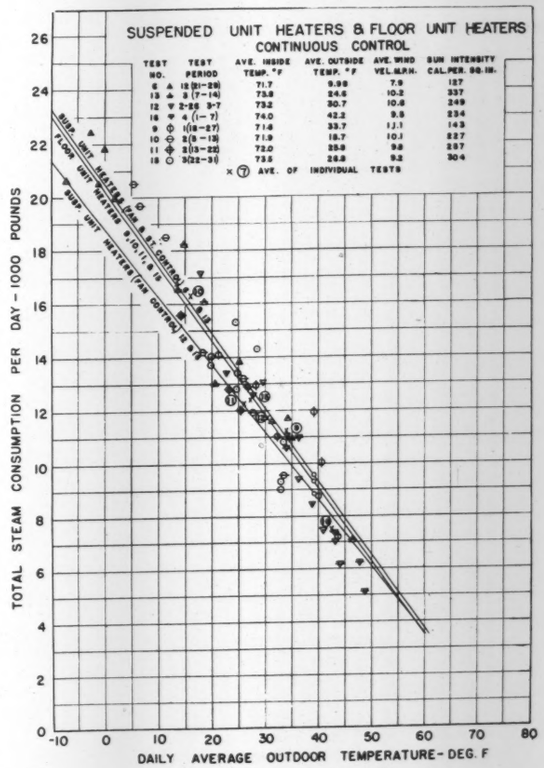
SUSPENDED UNIT HEATERS & FLOOR UNIT HEATERS
CONTINUOUS CONTROL
TOTAL STEAM CONSUMPTION PER DAY - 1000 POUNDS
DAILY AVERAGE OUTDOOR TEMPERATURE - DEG. F

Fig. 12—Steam consumption curves for suspended unit heaters and floor unit heaters with continuous control

to bring the temperature to 70 F from the night temperature of 52.5 F. The temperature continued to rise until 80 F was reached. Three additional heating periods were observed during the day, the temperature varying from 68 F to 77 F.

The suspended unit heater system required one-half hour to raise the temperature from 56.6 F to 70 F in the morning. The maximum temperature obtained in this initial heating period was 77.5 F. The steam came on five additional periods during the day. The variation was from 68-69 F to 75.5-76.5 F.

With the floor unit heaters the temperature rise was delayed about 25 min beyond the time of shifting from night to day thermostats. About 30 min was required for the damper to change from the fully closed to the fully opened position. After a temperature rise was indicated, in 15 min the temperature reached 70 F from the night temperature of 66-66.5 F. There were two heating periods during the day and the variation in temperature was from 70.5 F to 82 F.

The temperature gradient curves shown on Fig. 6 are the averages of a large number of readings on all days of the tests. The average temperatures from floor to ceiling are shown for each section and for each heating system. The curves of each section are consistent with each other except as to the location of the curves showing temperature of the floor unit heater tests in sections 2 and 3. These units were controlled by thermostats in sections 1 and 3. The two units were about 190 ft apart. As a result the temperatures from floor to ceiling ran lower for this system in the middle section than in the two end sections. In section number 3, temperatures above the 5 ft level were high for the floor unit heater system. The vertical line of resistance thermometers was 30 ft from the unit in this section and the readings were undoubtedly influenced by the direct discharge from the unit. Consequently the averages shown are higher than the true averages over the entire section number 3. This curve does show that on the average the heated air rises considerably above the discharge outlets of the unit heater. The outlets are 8½ ft high, whereas the maximum temperature 30 ft from the outlet was found 17 ft above the floor.

In each of the three sections the average temperature 1 ft above the floor was higher with the direct radiation system than with the two systems using forced air circulation. In all sections the temperatures at the 5 ft level checked closely for the direct radiator and suspended unit heater systems. In section 1, all three checked closely at this level. In section 3, the temperature at the 5 ft level for the floor unit heater was low, just as it was high above this elevation.

In section 1, the average temperature at 21 ft was slightly less than 77 F for direct radiation and was about 1 deg less for the suspended unit heaters and 2 deg less for the floor unit heaters. In section 2, the direct radiation system showed an average temperature at the 21 ft level about 2 F higher than did suspended unit heaters. Had the 5 ft level temperatures coincided, the floor unit heater system would have had a slightly lower ceiling temperature in this section as compared to that of the others as it did in section 1.

The noticeable dips in the temperature gradient curves for the 17 ft levels in sections 1 and 2 are considered due to the influence of skylights with ventilating perimeters. Table 3 indicates the spread of temperatures for these temperature gradient observations.

Conclusions

The following conclusions may be drawn as applying to the three heating systems and the conditions under which the tests were conducted in the Heating and Ventilating Laboratory:

1. For dual control with the day thermostat set at 70 F for 11 hours and the night at 50 F for 13 hours, the saving of steam over continuous control with the thermostat set at 70 F for 24 hours was:
   - (*a*) For direct radiation 15 per cent and probably a much larger saving had sun intensities and wind velocities been equal;
   - (*b*) For suspended unit heaters with fan and steam control 30 per cent;
   - (*c*) For suspended unit heaters with fan control and continuous steam 25 per cent.
2. For dual control with the day thermostat set at 70 F for 11 hours and the night at 60 F for 13 hours, the saving of steam over continuous control with the thermostat set at 70 F for 24 hours was:
   - (*a*) For suspended unit heaters with fan and steam control 24 per cent;
   - (*b*) For floor unit heaters, with mixing damper control, continuous steam and continuously running motors 16 per cent.
3. The saving of suspended unit heaters over direct radiation both operating on continuous control was something over 13.5 per cent, and might have been 20 per cent had sun and wind effects been equal.
4. The saving of suspended unit heaters over direct radiation both operating on dual control (70-50 F) amounted to 25 per cent.
5. The saving of floor unit heaters over direct radiation both operating on continuous control was 9 per cent and would have been higher had sun and wind effects been equal.
6. The saving of steam of floor unit heaters over suspended unit heaters both operating on continuous control was 2.3 per cent.
7. The saving of steam of suspended unit heaters over floor unit heaters both operating on dual control (70-60 F) amounted to 7 per cent.
8. The average temperature gradient for all days of tests for the three heating systems over the entire period of test resulted in a change of two-thirds of one per cent per foot of height above the 5 ft level.

Bon Voyage

After a visit of three weeks in the United States, Mr. and Mrs. J. Nelson Russell sailed for England, October 20, on the White Star liner Britannic. They enjoyed a motor tour through New England and the Catskill Mt. area and visited relatives on Long Island. Mr. Russell saw some of the outstanding heating and air conditioning installations in New York and discussed some of the technical phases of these subjects with American engineers. He addressed the New York Chapter at its October meeting and his fellow members of the A. S. H. V. E. enjoyed his discussion of British practice.

When the North German Lloyd liner Bremen sailed from New York, October 20, one of the passengers was Mr. Nikolai Alfsen, of Oslo, who is returning to Norway with many new ideas about American mechanical equipment and engineering practices observed during a six weeks trip through the principal cities of the East and Middle West, when he enjoyed meeting many of his fellow members of the Society for an exchange of ideas. He was greatly impressed with the Century of Progress and the extensive use of air conditioning in various types of American buildings.

Study of Unit Room Coolers in the Research Residence

By **A. P. Kratz*, M. K. Fahnestock**, and S. Konzo***** ***(MEMBERS)***

Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory and the National Warm Air Heating and Air Conditioning Association

The results presented in this paper were obtained in connection with the summer cooling investigations conducted during the summers of 1932[1] and 1933[2] in the Research Residence, Fig. 1, at the University of Illinois, by the Engineering Experiment Station under the direction of A. C. Willard, Acting Director of the Engineering Experiment Station and head of the Department of Mechanical Engineering. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Special acknowledgment is due the National Association of Ice Industries, the Utilities Research Commission of Chicago, Ill., the General Electric Co., the Carrier Corp., and the American Blower Corp. for their cooperation. Acknowledgment is also due E. L. Broderick, Research Assistant and A. F. Hubbard, Special Research Graduate Assistant, for active participation in the detailed work of the investigation.

Fig. 1—View of Research Residence in Urbana, Illinois

THE principal objects of this investigation were (1) the determination of the method of operation and the control of the unit room coolers, and (2) the determination of the performance of the unit room coolers as measured by their ice melting capacity and by their effect on the ambient air, including temperature, humidity, and motion.

A complete description of the Research Residence in which these tests were conducted is given in two previously published papers.[1, 2]

Description of Apparatus

Two types of commercial unit room coolers, Fig. 2, were used during these tests and their location in the living room on the first floor of the residence is shown in Fig. 3.

Unit *A* was a portable insulated ice chest, consisting of two compartments, one located above the other. The upper compartment was for ice storage, with a maximum capacity of 300 lb of ice when fully loaded, while the lower compartment formed a tank for holding the melted ice and water collected from the dehumidification of the air. The ice compartment was provided with special metal surfaces, in contact with the ice on one side and exposed to the air on the other; the side exposed to the air having extended fins which materially increase the heat transfer from air to ice. The warm air from the room, entering through a grilled opening in the front of the unit, Fig. 2, passed under and up the back side of the ice compartment in direct contact with the ice, and with the cold, finned, metal surfaces, into the suction inlet of the twin-motor centrifugal fans. The latter ejected the cooled air into the room from the top of the cooling unit. The water resulting from the melting of the ice, together with any condensation of moisture resulting from the dehumidification of the warm room air as it came into contact with the ice and cooling surfaces, was drained into the tank in the lower compartment.

For test purposes, this unit was mounted on a portable platform scale, which was sensitive to 0.01 lb, and both the ice meltage and dehumidification rates were determined by direct weights.

Unit *B*, Fig. 2, consisted of an insulated ice storage tank of 500 lb capacity, located in the basement, and a cooling unit placed in the living room on the first floor, Fig. 3. The cooling unit consisted of an attractively finished cabinet, which enclosed the extended surface cooling coils, dehumidification drip-pan, and fans. The cold water was pumped from the storage tank through the

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[1]Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo. A. S. H. V. E. Journal Section, *Heating, Piping and Air Conditioning*, February, 1933, p. 115.

[2]Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo. A. S. H. V. E. Journal Section, *Heating, Piping and Air Conditioning*, December, 1933, p. 613.

For presentation at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935.

cooling coils, over which the warm room air was circulated by means of a twin-motor centrifugal fan which could be operated at three different running speeds. For the tests discussed in this paper the lowest fan speed was used. The dehumidification, or moisture condensed out

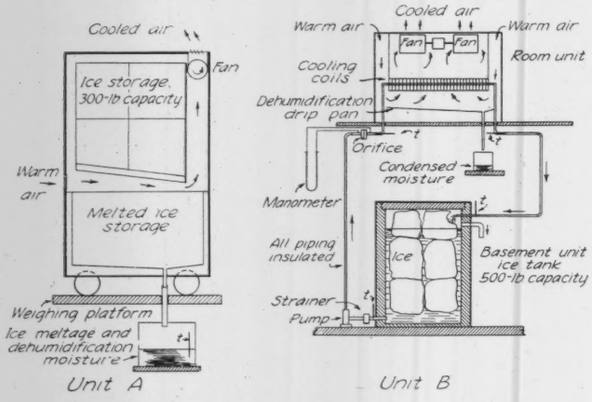

Fig. 2—Diagram of cooling units and test arrangement

of the air, drained from the cooling coils into a shallow drip pan beneath the coils from which it was collected and weighed.

It should be noted that both units *A* and *B* made no provision for introducing air from outside of the house, but merely recirculated the air in the room which was cooled.

The first floor of the Research Residence, Fig. 3, consisted of a living room, a hallway, a dining room, a breakfast nook, and a kitchen, with a total net volume of 7,300 cu ft. The living room, with a north, east, and south exposure was 13 ft 6 in. by 20 ft, with an 8 ft 11 in. ceiling, and had a total net volume of 2,410 cu ft. This room contained 6 windows, having a total exposed glass area of 90 sq ft. All of the windows in the house on the east, south, and west sides, which were exposed to the sun at some time during the day, were shaded with awnings, except in the case of a few tests when the awnings were raised for the purpose of determining their effect. The average occupancy during tests was one person per room.

Test Procedure

Operating Temperatures for Room Cooling Units: One of the most important factors involved in space cooling is the debatable question of the proper inside temperature to be maintained and its relationship to the outside temperature. Fig. 4 shows three schedules of indoor temperatures or methods of operating a unit room cooler. The schedule for the first method, designated as *Case* 1, was based on the assumption that the indoor air temperature should be maintained at a value 10 F less than the outdoor air temperature at all times. This condition is represented by the broken line, and the indoor air temperature actually maintained is shown by the solid line. It may be noted that with this schedule of operation it was necessary to start the cooler unit *A*, a considerable length of time before the maximum outdoor temperature was reached, at which time the indoor temperature conditions are indicated by point *b*, and that during the period indicated between the points *b* and *c*, the outdoor air temperature increased more rapidly than the indoor air temperature. It should also be noted that after the outdoor air temperature had reached a peak value and started to decline, it was necessary to progressively reduce the indoor air temperature in order to maintain the constant differential as specified by the schedule. This reduction in indoor air temperature, accompanied by a reduction in the temperature of the contents of the room, such as furniture, inside partition walls, etc., in the late evening hours when the cooling load was just reaching a peak due to the lag in heat transmission through the outside walls, imposed such an exceedingly heavy load on the unit as to make this schedule of operation impractical. A unit or plant of sufficient size to maintain this schedule in the late afternoon or evening would be prohibitive in first cost and uneconomical to operate. Furthermore, if the indoor conditions were conducive to comfort during the peak outdoor temperature, there was no indication or reason to believe that the maintenance of these conditions resulted in discomfort after the peak had passed.

In the method designated as *Case* 2, Fig. 4, an attempt was made to maintain the schedule of desirable indoor air temperatures recommended in THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' GUIDE, 1932.[3] This schedule provides for a variable differential between the indoor and outdoor temperature, the differential increasing or decreasing progressively as the outdoor temperature increases or decreases. The difficulties encountered with this method of operation were

[3] THE A. S. H. V. E. GUIDE, 1932, Chapter 2, p. 10, Table 2.

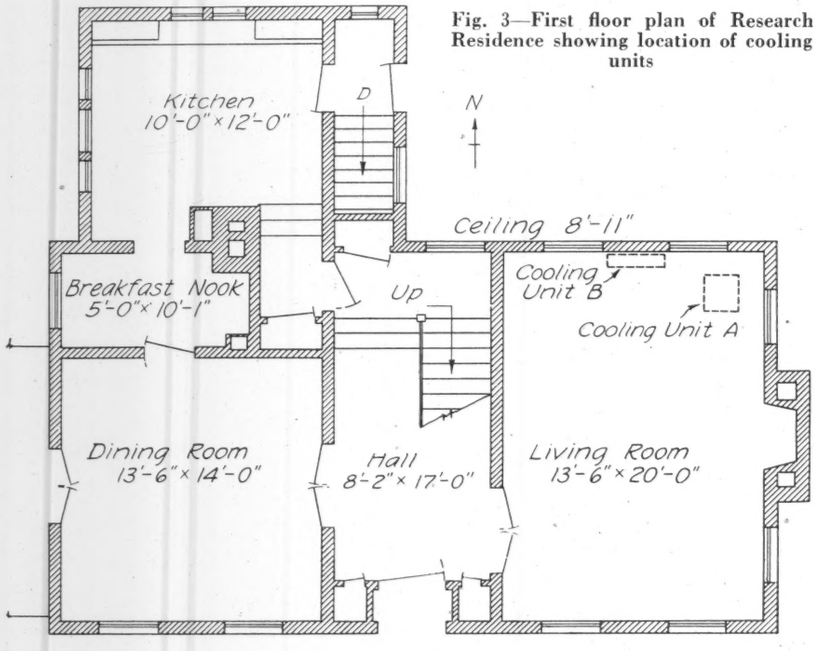

Fig. 3—First floor plan of Research Residence showing location of cooling units

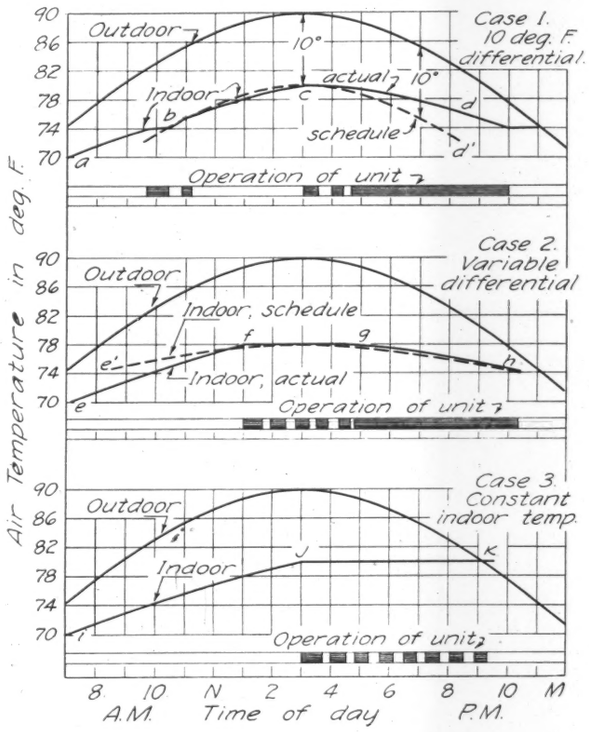

Fig. 4—Three schedules of room air temperature maintained with room cooling Unit *A*

similar to those already enumerated under *Case 1*, but somewhat less pronounced due to the variable differential. In addition, both methods required careful, impractical manual control and adjustment, which would be entirely unsatisfactory in a commercial or domestic installation.

The more simple method of operation for which the cooling plant was started when the effective temperature indoors reached some predetermined value, as for example 75 F, irrespective of the outdoor temperature, was finally adopted as a standard, and is designated as *Case 3* in Fig. 4. This usually occurred when the indoor dry-bulb temperature reached a value between 78 and 82 F, depending upon the relative humidity, although in some of the tests, slightly lower dry-bulb temperatures were maintained. The indoor dry-bulb temperature was maintained practically constant by operating the unit intermittently through the action of a simple two-point thermostat placed in the electric fan circuit. This method of operation was very satisfactory, and the indoor temperature conditions were maintained by operating the unit until the outdoor effective temperature dropped to approximately the same value as the indoor effective temperature, at which time the unit was stopped and the windows were opened.

The choice of a particular dry-bulb temperature to be maintained on a given day for any domestic air cooling unit installation is governed by a number of factors, including that of economy of operation. Obviously, for economical reasons, it is advisable to maintain the highest indoor air temperature conducive to comfort. For climatic conditions similar to those of Urbana, Illinois, experience has shown that an indoor dry-bulb temperature of approximately 81 F, with relative humidities between 40 and 60 per cent is very satisfactory, whereas for cooler climates it is probable that a slightly lower air temperature might be necessary. Regardless of the particular indoor air temperature to be maintained, there seems to be little need for complex schedules of operation such as those illustrated by *Cases 1* and *2*.

Test Methods Used with Unit Cooler A: As previously stated, unit cooler *A*, Fig. 2, was mounted on a portable platform scale which was sensitive to 0.01 lb, and both the ice meltage and dehumidification rates were determined by direct weighing. At the start of each test the unit cooler was charged with ice, thoroughly pre-cooled, the water drained out, and the gross weight of the unit and ice was observed. It was then operated according to a predetermined temperature schedule, *Case 3*, Fig. 4, and at the end of each hour the ice meltage and dehumidification were drained directly into a separate container and weighed as shown in Fig. 2. At the same time, after the water had been drained out, the gross weight of the unit and ice was observed, and the net ice meltage was obtained from the difference in these hourly readings. Since the water drained from the unit each hour consisted of both the ice meltage and the moisture condensed out of the air, the net amount of moisture condensed, or the dehumidification, was determined from the difference between the total weight of water collected and the net weight of ice melted. By this method of procedure both the hourly rates of ice meltage and dehumidification were very satisfactorily obtained without handling the ice or the unit.

The majority of the tests with this unit were made with the unit located in the living room of the residence, Fig. 3, but several tests were also made with it located in the dining room or in the hall. When located in the living room or in the dining room, the doors between that room and the hall were closed, confining the cooling entirely to the one room. When located in the hall, the doors to the living room and dining room were opened and the entire first story was cooled.

Test Methods Used with Unit Cooler B: The amount of heat absorbed by the cold water as it passed through the extended surface cooling coil in unit *B*, Fig. 2, could be determined by obtaining the weight of water circulated and the temperature rise of the water through the coil. From Fig. 5, it may be observed that although the temperatures of the entering and leaving water responded very quickly to the operation of the fan, there was sufficient lag in these temperatures after the fan was started and stopped to make it difficult to accurately determine their average values. Since the total temperature rise was of such small magnitude that a slight variation in either temperature resulted in a comparatively large variation in the temperature rise, and hence in the computed heat quantities, this method of measuring the heat absorption was replaced by the more direct method of weighing the ice melted during a test.

Previous to the start of a test the water in the tank and system was precooled. The weight of the initial charge of ice, and subsequent charges during a test were then accurately observed. At the end of a test the amount of ice remaining in the tank was deducted from the total amount charged, giving the net amount used in cooling and dehumidifying. As previously stated the moisture condensing on the cooling coil was drained into a container and weighed directly.

All of the tests with unit *B* were conducted with the cooling section of the unit located near the north wall of the living room, on the first floor of the Research Residence, as shown on the floor plan in Fig. 3. During some of the tests the door from the living room to the interconnecting hallway was opened and the unit allowed

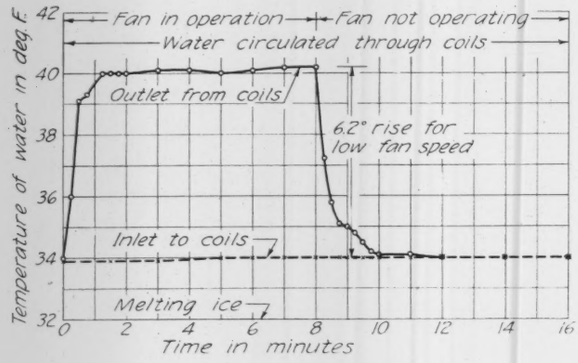

Fig. 5—Typical temperature rise of cooling water in Unit *B*

to cool the entire first story, consisting of the living room, the hallway, the dining room, the breakfast nook and the kitchen, having a total net volume of 7,300 cu ft.

As with unit *A*, unit *B* was operated on a predetermined temperature schedule, *Case 3*, Fig. 4, maintaining a fixed dry bulb temperature. It was operated on the *on* and *off* principle and was controlled by means of a room thermostat connected into the electric circuit to the fan.

Test Results

Performance of Unit Cooler A: A typical performance record for unit *A* when cooling the living room only is shown in Fig. 6. In this test the unit was initially charged with 206.0 lb of ice and the dry bulb temperature of the air in the living room was maintained at a constant value of approximately 75.5 F. The test was terminated at the end of 13.5 hours when the 206.0 lb of ice were all melted, and it was merely a coincidence that at this time the effective temperature outdoors was approximately the same as the effective temperature in the living room. It is significant to note however, at this time, that the unit could have been stopped and cooling continued by the introduction of night air even if the ice supply had not been depleted.

The average dry bulb and wet bulb temperatures of the air entering and leaving the cooling unit during the periods of fan operation are given:

	Entering Air	*Leaving Air*
Dry bulb temperature	75.0 F	63.8 F
Wet bulb temperature	64.3 F	58.0 F
Dewpoint temperature	58.6 F	54.4 F
Relative humidity	56.0 per cent	71.0 per cent
Moisture content of the air	73.57 grains per lb of dry air	62.97 grains per lb of dry air

It should be noted that the air was cooled 11.2 F in passing through the unit and that the moisture content was reduced 10.6 grains per pound of dry air.

The dry bulb temperature of the air in the living room was maintained at a level approximately 2 F lower than that in the uncooled hall due to the operation of the unit. It should also be noted that a comparable cooling effect was obtained by the reduction in the relative humidity. From Fig. 6 it may be observed that at 10:30 a. m. when the unit was started, the relative humidity of the air in the room was 68 per cent; at noon it had been lowered to 55 per cent and between that time and 8:00 p. m. it ranged from 51 to 55 per cent. In cooling effect this

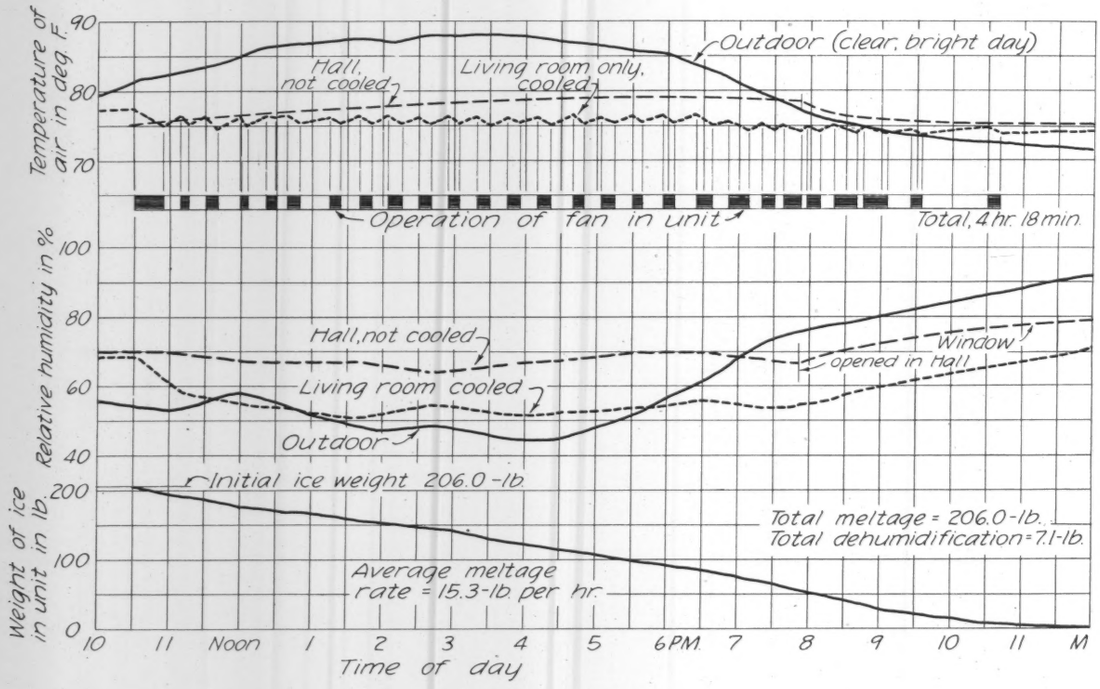
Temperature of air in deg F
Outdoor (clear, bright day)
Hall, not cooled
Living room only, cooled
Operation of fan in unit
Total, 4 hr. 18 min.
Relative humidity in %
Hall, not cooled
Window opened in Hall
Living room cooled
Outdoor
Weight of ice in unit in lb.
Initial ice weight 206.0-lb.
Total meltage = 206.0-lb.
Total dehumidification = 7.1-lb.
Average meltage rate = 15.3-lb. per hr.
Time of day

Fig. 6—Temperature and humidity data for test with room cooling Unit *A*. (Living room only cooled; doors between living room and hall closed)

reduction of approximately 15 points in relative humidity is equivalent to a reduction of 1.5 deg in effective temperature or to a 2.0 F reduction in dry bulb temperature.

The total dehumidification, or the weight of moisture condensed from the air during the period when 206.0 lb of ice were melted was 7.1 lb. This change in the moisture content of the air was equivalent to a heat absorption of approximately 7,474 Btu. The total heat absorbed by the ice melting process was composed of the latent

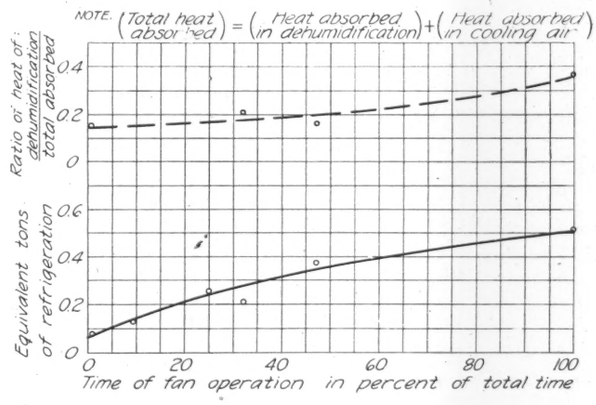

Fig. 7—Performance curves for Unit Cooler *A*

heat of fusion of the ice, plus the sensible heat absorbed by the water resulting from the meltage and was equal to

$206.0 \times 144 + 206.0\ (57\text{-}32) = 34{,}814$ Btu.

144 = latent heat of fusion of ice

57 F = temperature of water going to the drain.

Of the total amount of heat absorbed by the ice melting process, during the entire period of the test, approximately 21.5 per cent was given up by the change in the moisture content of the air, consisting of the total heat given up by the vapor condensed and the change in superheat of the vapor remaining in the air. As shown by the upper curve in Fig. 7, the amount of dehumidification accomplished with this type of cooling unit was dependent on the percentage of the time that the fan in the unit was operating. For example, the ratio of the heat absorbed in the dehumidification process to the total heat absorbed, varied from a value of 37.8 per cent when the fan in the unit was operating continuously, to a value of approximately 16.0 per cent when the fan was not running and during which time the small amount of ice meltage was that due to standby heat leakage only.

The average rate of ice meltage, as indicated by the lower curve in Fig. 6, was approximately 15.3 lb per hour for a period of 13.5 hours. The maximum meltage rate observed with the fan operating continuously was 43.8 lb per hour, with a room temperature of about 75 F.

The values of heat absorption may be reduced to an equivalent ice melting capacity expressed as tons of refrigeration by dividing the heat absorption rate in Btu per hour by 12,000. Therefore, for the test shown in Fig. 6, the total heat absorption of 34,814 Btu for the 13.5 hour period was equivalent to 0.215 tons of refrigeration. The summary of results from six tests is presented in Fig. 7 and it may be noted from the lower curve that the ice melting capacity expressed in tons of refrigeration varied from a value of 0.51 tons with continuous fan operation to a value of 0.06 tons when the fan in the unit was not running.

Performance of Unit Cooler B: The performance of unit cooler *B* was similar to that of unit *A*. However, it was possible to obtain a wider variation in cooling rates with unit *B* by making adjustments in the rate of circulation of the cooling water, by changing the speed of the fan or by allowing the fan to run a larger proportion of the time. In the tests reported in this paper the rate of circulation of the cooling water and the fan speed remained constant. The amount of cooling depended entirely upon the length of time that the fan in the unit was operated, and Fig. 5 shows the sensitivity and positiveness of this method of control as reflected in the temperature rise and fall in the cooling water, corresponding to the operation of the fan. Approximately one minute after the fan started or stopped the temperature of the water leaving the cooling coils had practically reached equilibrium. For the test shown, the temperature of the water entering and leaving the coils was approximately 34 F and 40 F respectively, giving a temperature rise of about 6 F through the coils.

Fig. 8 shows a typical performance curve with the unit located in the living room as shown in Fig. 3, and cooling the entire first story consisting of the living room, hall, dining room, breakfast nook and kitchen; or a total volume of 7,300 cu ft. It may be noted that the outdoor temperature reached a maximum value of 93.7 F at about 3:30 p. m. and that the temperature in the southwest bedroom on the uncooled second floor, attained a maximum value of 87.5 F at about 7:30 p. m. During the test period of 11.67 hours it was possible to maintain average breathing level temperatures of 79.3 F in the living room, 81.5 F in the hall, 82.6 F in the dining room, and 82.1 F in the kitchen, or an average of 81.4 F for the entire first story. Although the cooling unit and control thermostat were unfavorably located with respect to the center of the first story, the air temperatures in any one room did not deviate more than 2.1 F from the mean temperature for the entire first story. The minimum room temperature of 79.3 F occurred in the living room where the cooling unit was located, while the maximum room temperature of 82.6 F occurred in the dining room. Unquestionably, this variation in air temperature between rooms would have been reduced by a more central location of the thermostat and unit. It is also probable that, from the standpoint of temperature equalization in several rooms, the best method of operation for a unit room cooler of this type is that which will allow the unit to operate for the greatest portion of the time, at a rate of cool air delivery just sufficient to balance the heat load. With such a unit equipped with a multiple speed fan this method of operation can be most closely approximated by running the fan at the lowest available speed which will handle the load; whereas, in a single speed fan unit the adjustment could be made either by employing adjustable louvers to the amount of air delivered, or by using a control valve to vary the rate of water being circulated through the cooling coils.

The total ice meltage for the test shown in Fig. 8 was 500 lb, and the total amount of heat absorbed was 82,224 Btu, which included the net rise in the sensible heat of the water in the ice tank.

The average rate of heat absorption reduced to an equivalent ice melting capacity and expressed as tons of refrigeration was 0.59 tons. In general, the tests with both units *A* and *B* on the first story indicated that a cooling unit with a capacity of approximately 0.25 ton of refrigeration per room would be required to handle the maximum cooling requirements of rooms similar to those in which these tests were made, and in which no

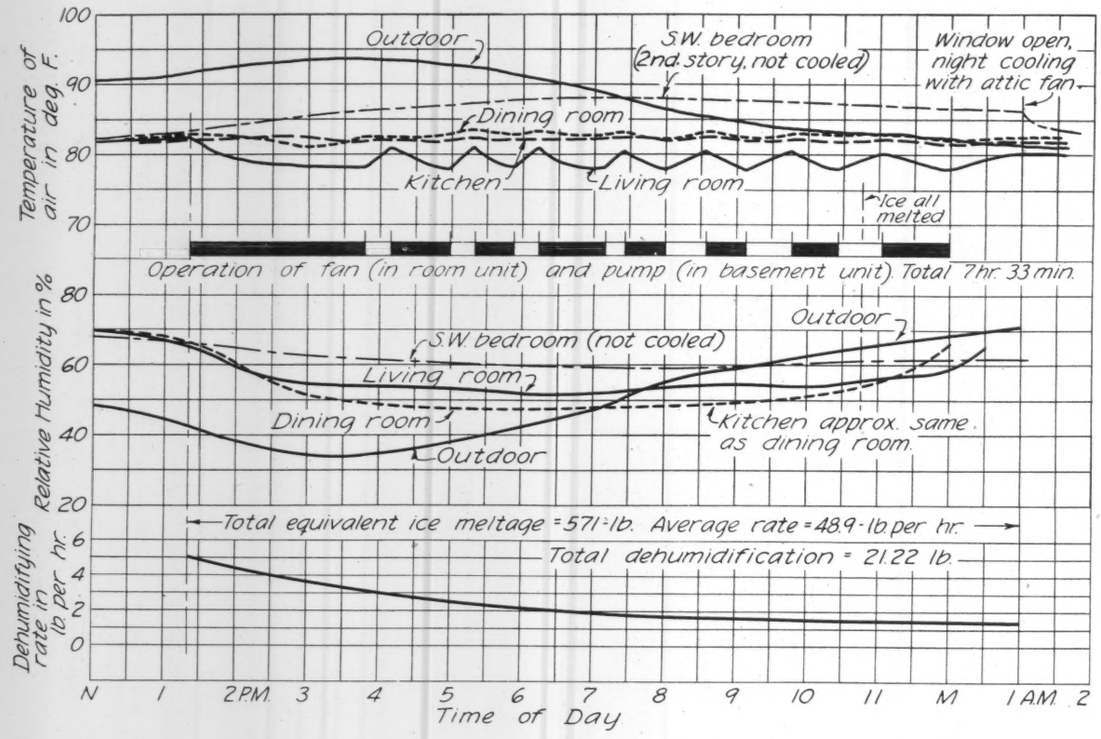

Fig. 8—Temperature and humidity data for test with room cooling Unit *B*. (Entire first story cooled; all doors between rooms opened)

outdoor air other than that which occurs by natural infiltration was introduced for the purpose of ventilation. It should be noted that the windows of these rooms which were exposed to the sun were equipped with awnings, and that the occupancy load consisted of only one person per room. There was practically no power load. The estimate of 0.25 ton of refrigeration per room is based on normal running conditions, since starting conditions impose an extra load of considerable magnitude. It is very probable that a larger tonnage capacity per room would be necessary on second story rooms since the cooling load for the second story is considerably greater than that for first story and the cooled air would tend to drift downwards to the first story.

The total dehumidification or amount of moisture condensed out of the air during the entire test lasting 11.67 hours was 21.22 lb, and the ratio of the heat absorbed by the change in the moisture content of the air to the total heat absorbed was 27.2 per cent. The large magnitude of the dehumidification load is emphasized by the lower curve in Fig. 8, showing the rate of dehumidification, or the weight of water condensed out of the air in pounds per hour. It may be observed that the initial rate at the beginning of the test was approximately 5.0 lb per hour, while the average rate for the entire test period was only 1.82 lb per hour. As shown by the center group of curves in Fig. 8, this drying process substantially reduced the relative humidity of the air in all of the rooms on the first story. The average initial relative humidity of approximately 68 per cent was reduced to a value of 58 per cent at the end of one hour's operation and at the end of five hours' operation, it had attained a value of about 50 per cent. Since the dry bulb temperature remained constant the reduction in relative humidity also represented a reduction in moisture content, or the pounds of moisture present per pound of the air. The fact that during the operation of the cooling unit the moisture content on the second story remained constant indicated that the humidities in the rooms on the second story were also somewhat reduced by the dehumidification accompanying the cooling on the first story, since it was observed on other days that when the cooling unit was not operated a gradual increase in the moisture content of the air on the second story always occurred during the day.

Summary and Conclusion

General Summary: Although these tests indicate that a unit room cooler can be successfully adapted to cool from one to three rooms on the same floor of a residence, it is probable that the most economical method of cooling would be to operate such units in conjunction with a fan unit for the purpose of introducing cool night air into the rooms or house. It would be feasible, for instance, to locate the cooling unit on the first story of a two story

house and operate it during the afternoon and evening, while the second story rooms would be cooled at night by introducing night air through the open windows.

The advantageous features of portability and lower initial cost of a unit similar to unit *A* are somewhat offset by the inconvenience of having the entire unit in the living quarters. It would probably be preferable to locate the unit in the basement and attach a short duct system from the outlet of the unit to a register on the first story, thereby avoiding the necessity for servicing the unit in the living portion of the house.

Adequate insulation of the ice storage compartment in all types of units is very essential for the minimization of standby losses. The cooling load during the summer was extremely variable with the weather conditions fluctuating rapidly, and oftentimes it was found that after the ice storage tank was filled, very little cooling was necessary due to sudden temperature changes outdoors. Under these conditions standby losses become expensive if any appreciable ice meltage takes place between operations. With units similar to type *B*, it is essential that all pipes carrying relatively cold refrigerants be thoroughly insulated.

The matter of noise of mechanical parts and of air motion is extremely important, and great care should be taken in the design and installation of any unit to reduce these features to a minimum.

Conclusions: The following conclusions are applicable to the conditions under which these tests were made:

1. Unit room coolers should be operated and controlled to maintain a constant dry bulb temperature in the room, rather than to maintain a complex schedule of indoor temperatures varying with the outdoor temperature.

2. Unit room coolers, similar to units *A* and *B*, can be successfully adapted to cool from one to three medium sized rooms on the same floor of a residence.

3. A unit cooler of 300 lb ice capacity and having a melting rate of approximately 20 lb of ice per hour would be adequate to handle the normal cooling requirements of a medium sized first story room of a residence for most summer days, if no outdoor air other than that which occurs by natural infiltration is introduced for ventilation. However, if the occupancy of the room is to be more than two persons it is essential that additional capacity be supplied for the necessary ventilation required to reduce undesirable odors.

4. The initial ice melting rate, or the melting rate during the first hour after the unit was started, was nearly twice the value of the normal melting rate.

5. Of the total amount of heat absorbed by the unit coolers, approximately from 16 to 38 per cent was absorbed in the dehumidification process.

6. In addition to the lowering of the dry bulb temperature the relative humidity of the air in the rooms was, in most cases, reduced about 15 points after three or more hours of continuous operation of the units.

7. Although not centrally located on the first story of the residence, the cooling units operated in the living room effectively cooled and dehumidified the air in the adjoining rooms and there was some indication that the reduction in relative humidity extended to the second story.

8. With unit cooler *B* located in the living room and all interconnecting doors on the first story open, the maximum deviation of the breathing level temperatures in any room on the first story from the mean temperature of the entire first story was 2.1 F.

Congratulations, Colonel

A military wedding at Framingham, Mass., on October 10, completely surprised the many friends of Lt. Col. Davis S. Boyden and Miss Mary Abie Doolan. The marriage ceremony was performed by Brig. Gen. O'Connor (retired), Massachusetts National Guard.

Having returned from a honeymoon in the White and Green Mountains, Colonel and Mrs. Boyden are receiving the hearty congratulations of their friends.

Second Edition of Refrigerating Data Book Published

The *American Society of Refrigerating Engineers* announces its publication of the Refrigerating Data Book, second edition, 1934-36, a reference handbook and catalog for refrigerating engineers. The book contains 486 pages of comprehensive information on all subjects which are of interest to the refrigeration engineer and others in the industry. Its five principal sections, which cover fundamental data, principles, refrigerating machinery, refrigerating equipment, and applications of refrigeration, are each divided into several chapters which illustrate and amplify the topics considered in an authoritative and concise manner.

Besides presenting the compilation of many data prepared by its large staff of experienced men, the book shows, by means of well-selected bibliographies, where supplementary information and discussion may be found. Only approximately one third of the first edition has been reprinted, some 48 pages have been added, and 17 out of 30 chapters are new or wholly revised. A comprehensive index, a catalog data section of manufacturers' equipment, and a list of *A. S. R. E.* members complete the book.

The editorial policy of the original Data Book Committee has been continued and the scope of treatment proceeds from an estimate of the needs of the reader who is assumed to have a knowledge of engineering. The personnel of the Committee is:

Alvin H. Baer	Glenn Muffly
Harry D. Edwards	Alfred W. Oakley
Crosby Field	A. R. Stevenson, Jr.
Harry Harrison	H. M. Williams
Louis H. Morse	Frank R. Zumbro

David L. Fiske, *Editor*

Principles are well treated in the book which is fully illustrated by charts and figures of excellent quality. By using examples, many of a graphic nature, to emphasize the highlights in the different chapters, the editors believe that the book has been made useful for general reference and for the use of the student. The publishers, have, by means of footnotes, given the reader a full and accurate idea of the sources of the material used. They have also included a very fine system of references and cross-references, which are of great assistance.

A full and complete listing of abbreviations, symbols, and conversion data in the front part of the Data Book is convenient, and in the back a refrigerating glossary defines the many technical expressions and terms found in the text.

Improved and revised characteristic charts and tables for the various refrigerants used in the refrigerating systems are an important improvement in the new edition and constitute the basis for a new chapter. Among other entirely new chapters in the edition are: Thermodynamics of Refrigerating Cycles, the Ammonia Absorption System, Heat Transmission in Refrigeration, Small Condensing Units, and Brewery Refrigeration.

An attractive embossed grained leatherette cover binds the book, which may be secured from the publishers, the *American Society of Refrigerating Engineers,* 37 West 39th St., New York, N. Y.

NOMINATIONS FOR 1935

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1935, submits the following list of nominees:

For President:

JOHN HOWATT, Chicago, Ill.

For First Vice-President:

G. L. LARSON, Madison, Wis.

For Second Vice-President:

D. S. BOYDEN, Boston, Mass.

For Treasurer:

A. J. OFFNER, New York, N. Y.

For Members of the Council:

Three-Year Term

J. F. MCINTIRE, Detroit, Mich.
ALBERT BUENGER, St. Paul, Minn.
F. E. GIESECKE, College Station, Texas
W. E. STARK, Cleveland, Ohio

Respectfully submitted,

NOMINATING COMMITTEE.

W. R. EICHBERG, *Chairman;*
W. G. BOALES, *Acting Secretary.*

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. B-VIII—Section 10. The Nominating Committee shall consist of one (1) member eligible to vote designated by each Chapter, or his alternate also appointed by the Chapter. The Secretary of each Chapter shall certify to the Secretary of the Society on or before January first the names of the member and alternate selected.

The Committee shall meet at the Annual Meeting of the Society at the call of the Secretary of the Society and shall effect its own organization and elect its own Chairman. At the Semi-Annual Meeting of the Society, if possible, the Nominating Committee shall select the nominees for the ensuing year for the offices of President, First Vice-President, Second Vice-President, Treasurer, and four (4) members of the Council. In any event the names of the nominees shall be certified to the Secretary of the Society before September twentieth, with the written consent of each nominee to fill the office for which he has been selected and their names with the offices to which they have been nominated shall be published in the October issue of the JOURNAL.

Art. B-IX—Section 2. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, eligible to vote, at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election.* The election is governed by the By-Laws for the election of officers, with the single exception that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the Government of the Research Laboratory, adopted at the January, 1919, Meeting, and amended January, 1933, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1935:

Three-Year Term

C. A. DUNHAM, Chicago, Ill.
W. L. FLEISHER, New York, N. Y.
ELLIOTT HARRINGTON, Schenectady, N. Y.
A. P. KRATZ, Urbana, Ill.
H. C. MURPHY, Louisville, Ky.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

ARTICLE II—ORGANIZATION

Section 1. *Committee on Research*—There shall be a standing committee known as the Committee on Research, consisting of fifteen (15) members each serving for three (3) years and five (5) retiring each year. The outgoing Chairman if serving in that capacity during the last year of his three (3) year term on the Committee shall without election become an additional member of the Committee on Research for one (1) year.

(a) The Council shall nominate previous to July first of each year five (5) members to fill the vacancies of those retiring at the next Annual Meeting.

(*b*) The nominations made by the Council shall be published in the October issue of the Society's Journal.

(*c*) Any ten (10) members of the Society eligible to vote may present to the Secretary over their signatures, the name of one (1) or more additional nominees for the Committee on Research, provided such name or names are presented at least sixty (60) days prior to the next Annual Meeting, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

(*d*) The election shall otherwise conform to the regulations provided for the election of officers of the Society in the Constitution, By-Laws and Rules.

(*e*) Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

41st Annual Meeting

New City Hall, Buffalo

BUFFALO has been selected for the Society's 41st Annual Meeting and the members of the Western New York Chapter are making extensive plans for the pleasure and the enjoyment of their fellow members, who are cordially invited to be present at Hotel Statler, January 28-30, 1935.

These dates may seem to be far away but plans should be made now. Save some of that Christmas Fund and budget it for the trip to Buffalo. While the men are in the technical sessions there will be fun for the ladies. Then there will be evening parties for everyone.

Peace Bridge connecting the United States and Canada

Niagara Falls—Rock of Ages and Cave of the Winds Bridge

Something unique is being prepared by the Committee on Arrangements, of which M. C. Beman is chairman and Roswell Farnham is vice-chairman. Walter Voisinet says the banquet will be different—there will be no speeches but a good time for everyone. L. A. Harding, past president of the Society, heads the greeter committee and J. J. Landers will supervise the Registration. Cecil Farrar, chairman of Publicity, says he will tell everyone about Buffalo and Mr. and Mrs. Park Hedley will see that the ladies are entertained. D. J. Mahoney says it will be a grand party, C. A. Evans, director of Transportation, will show the sights, and B. C. Candee warns that there must be no deficit while he is chairman of Finance.

At a meeting of the Council in Buffalo, October 22, the technical program for six sessions was approved. Papers selected by the Program and Publication Committees include the following subjects:

Heat Requirements of a Building, by J. H. Walker and George Tuttle

Some Recent Studies on the Ashes in Coal, by Prof. T. G. Estep.

Insulating Value of Double Glass, by R. A. Miller

Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and J. James

Study of the Room Cooling Load in an Office Building, by F. C. Houghten and Carl Gutberlet

The Nature of Ions in Air and Their Possible Physiological Effects, by Prof. L. B. Loeb

Ventilation of Hospital Operating Rooms, by R. B. Saxon

Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, M. K. Fahnestock, S. Konzo and E. L. Broderick

Study of Unit Room Coolers in the Research Residence, by A. P. Kratz, M. K. Fahnestock and S. Konzo.

The cost of traveling by train is low, the round trip fare being on the basis of one and one-third fares.

Members will receive identification certificates from Society Headquarters by January 10 which can be presented to local ticket agents for the purchase of low rate round trip tickets to Buffalo.

For example—New York or Philadelphia members all expense rate of $48 to $50 will cover: (1) Round trip railroad fare and Pullman lower berth; (2) Single room and bath in Hotel Statler and meals (Mon. to Wed.); (3) Taxi and tips; (4) Banquet ticket and Get-together Party. Members from Detroit, Cleveland or Pittsburgh will find the cost materially less while those from Baltimore, Washington, St. Louis and Kansas City will have a little greater railroad expense.

Buffalo is a delightful city and many members will have pleasant recollections of the 1926 meeting. The headquarters hotel is located at Niagara Square directly across from the new City Hall. This is Buffalo's civic center with the McKinley Monument in the center.

The city of Buffalo occupies a key position on the Niagara Falls frontier and has much of historic interest for the visitor.

Annual Reports of Local Chapters

Kansas City

The annual report of the Kansas City Chapter has been prepared by Secy. L. R. Chase and the following officers are in charge:

President—L. A. Stephenson
Vice-President—C. A. Weiss
Secretary—L. R. Chase
Treasurer—G. L. Bliss
Board of Governors—David Caleb, C. A. Flarsheim, Harry Filkins

At the present time 52 members are on the Chapter roll.

A meeting was held on September 11, 1933 with an attendance of 16 members and 2 guests. The principal speakers of the evening were Messrs. Hutchinson and Yost, who gave a talk on acetylene welding, illustrated by motion pictures, and by an actual welding demonstration.

The October 5 meeting was held with 21 members and 4 guests present. The guest of honor was W. T. Jones, president of the A. S. H. V. E., who discussed with the Chapter problems of the Society for the next year.

On November 7 a meeting was held with 25 members and guests in attendance. The principal address was given by F. D. Berkeley on Steam Jet Refrigeration.

A meeting was held on December 11 with an attendance of 24 members and 6 guests. L. B. Miller spoke on Effective Temperature Control.

On February 19 the program consisted of a report on the Annual Meeting by chapter members who had attended. Nineteen members were present.

Hugh O'Brien addressed the March 19 meeting, when 23 members and 4 guests were present, on the Problems and Accomplishments of Air Transportation.

At the April 9 meeting a discussion on the relative advantages of using spray dehumidifiers versus surface cooling coils in air conditioning work was given by L. R. Chase, who spoke on dehumidifiers, and Sam Shure, who described the use of surface cooling coils. Attendance—23 members and 4 guests.

C. V. Haynes, president of the A. S. H. V. E., spoke on Your Society—What Does It Mean to You at the May 10 meeting, when the present officers were elected. Attendance—26 members and 9 guests.

Five new members were added to the Chapter roll during 1933-34.

During the year the Chapter sponsored an Air Conditioning Forum, which presented six meetings to the general public of Kansas City and the surrounding territory for their education with regard to air conditioning. The committee in charge of this work consisted of Carl Clegg, E. K. Campbell, and L. A. Stephenson. Due to the fact that the Chapter's By-Laws were very much antiquated, they were revised by a committee under the supervision of E. K. Campbell and were adopted by the Chapter.

Massachusetts

The Papers and Meetings Committee of the Massachusetts Chapter has tentatively scheduled the following meetings for the next few months; November—Reception to C. V. Haynes, president of the A. S. H. V. E.; December—Address by Mr. Wilson of the Babson Institute on Current Economic Conditions; January—A talk by C. W. Colby on the oil burning installation at Mt. Holyoke College.

The Massachusetts Chapter has 44 active members and the officers and committees are:

President—R. S. Franklin
Vice-President—J. W. Brinton
Secretary—W. A. McPherson
Treasurer—E. W. Berchtold
Board of Governors—Leslie Clough, L. A. Brissette, G. B. Gerrish, C. T. Flint, A. B. Jackson, and the officers

COMMITTEES

Meetings and Papers
J. W. Brinton, *Chairman*

Social
C. T. Flint, *Chairman*
C. R. Swaney
A. A. Klonower

Membership
L. A. Brissette, *Chairman*
I. S. Dane
W. F. Gilling
N. J. H. Shaw

During 1933 and 1934 the following meetings were held:

Meeting	*Speakers*	*Subject*
October 9	H. B. Smith H. E. Hackensmith J. W. Brinton	Air Conditioning for Residences
November 13	W. S. Parker, W. H. Oleson, E. W. Smith	National Building Congress
December 12	A. L. Jaros R. S. Franklin	Panel Heating
January 8		Reception to National Officers
February 26		Special meeting on revision of By-Laws
March 12	W. W. Wells Philip Drinker	Methods of Estimating Dust Particles and Bacteria in Air
April 9	R. T. Kern	Forced Hot Water Heating
May 1	W. T. Jones	Installation of Officers
May 14	C. T. Flint, E. A. Dusossoit, W. Bertillion, E. Laube	Symposium on House Heating

October 8, 1934. The regular monthly meeting was held in Durgin and Parks Restaurant when dinner was served at 6:30.

Later J. F. Malley, State Director, spoke for several hours on the Federal Housing Administration, and his talk was supplemented by several reels of motion pictures. Many questions followed Mr. Malley's discussion and evidenced the interest of those present in this subject.

Ontario

The secretary of the Ontario Chapter, H. R. Roth, in his annual report has outlined plans for the first three meetings of the Ontario Chapter, to be held at the Royal York Hotel. At the October 1 meeting, five local members spoke on various types of fuel and combustion, including gas, oil, underfeed stokers, etc. Heating and Ventilating Research will be the subject of the November 5 meeting when Mr. Ellis of the Ontario Research Laboratory will be the speaker. Arrangements are being made to have a member of the *American Gas Association* discuss Air Conditioning at the December 3 meeting.

The officers and committees for the years 1934-35 are:

President—W. R. Blackhall
Vice-President—M. W. Shears
Secretary-Treasurer—H. R. Roth
Board of Governors—J. S. Paterson, Thomas McDonald, G. A. Playfair and W. P. Boddington
Membership Committee—Thomas McDonald, O. L. Maddux, D. O. Price.
Attendance Committee—M. W. Shears, M. W. Best, A. H. Hills.
Entertainment Committee—G. A. Playfair, H. B. Jenney, H. H. Angus

The Ontario Chapter has 59 members enrolled and during the past season meetings were held at the Royal York Hotel except in May, when the members assembled at the Engineers Club. A synopsis of the meetings follows:

October 2, 1933. Speaker—M. W. Shears, Toronto, on The Continental System of Low Pressure Boiler Protection. Attendance—24.

November 6, 1933. Speaker—R. J. Thompson, on Freon as a Refrigerant. Attendance—70.

December 4, 1933. Speaker—A. S. Leitch on The Development of Heating and Ventilation During the Past 30 Years. Attendance—29.

February 6, 1934. Speaker—Mr. Storey on Engineering in Relation to Public Health. Attendance—38.

March 5, 1934. Speaker—H. H. Angus, Toronto, on Engineering Problems in the Design of Hospitals. Attendance—44.

April 2, 1934. Speaker—E. L. Hogan, on Air Conditioning. Attendance—64.

May 15, 1934. Speakers—C. V. Haynes, president of the Society, on What the Society Means to Its Members; John Howatt, 1st vice-president of the A. S. H. V. E. and chairman of the Committee on Research, on Research Activities of the Society in Pittsburgh. Attendance—37.

Pacific Northwest

Under the able direction of P. M. O'Connell, the Pacific Northwest Chapter enjoyed a successful season in 1933 and 1934. During that period the Chapter obtained 8 new members and it is hoped that the membership will be increased by a greater number during the coming months.

The October, 1933, meeting was well attended and the Chapter was honored to have W. T. Jones, president of the A. S. H. V. E., as speaker. Mr. Jones' talk was an inspiration to all present.

A dinner at the La Tosca Cafe preceded the November meeting when the speaker of the evening was Mr. Paulson, who described a boiler burner combination and air conditioning unit. E. L. Weber, consulting engineer, spoke on Panel Heating, a subject on which he is well informed and which proved of interest to the members and guests.

On December 19, twenty-four members and guests heard Prof. S. K. Kirsten of the University of Washington talk on the advancement of neon lighting. During the latter part of the meeting, under the able direction of A. L. Pollard, an inspection was made of a new 600 hp electrical high pressure boiler installed as a part of the central steam heating plant.

A joint meeting with the local section of the *A. S. M. E.* on January 19, 1934, was devoted to the subject of air conditioning. The speakers of the evening were E. L. Weber and Harry Brod.

In February a visit was made to the plant of the Century Brewing Co.

New members of the Chapter addressed the meeting in March and D. C. Griffin and K. F. MacLeod discussed articles of interest in current publications.

F. J. Pratt, of the machinery drafting department of the Bremerton Navy Yard, was present at the May meeting and described in detail ventilating problems aboard modern battleships.

In May the following members were unanimously elected and installed as Chapter officers:

President—A. L. Pollard.
Vice-President—Lincoln Bouillon.
Secretary—S. D. Peterson.
Treasurer—W. W. Cox.

Secretary Peterson reports that in July a summer meeting was held when members of the Chapter and their wives spent a weekend at the site of the municipal power development.

Philadelphia

Arrangements for the first three meetings of the Philadelphia Chapter, to be held at the Engineers Club, have been announced and on October 11, 1934, T. N. Adlam presented a paper on Panel Heating.

On November 8, the subject selected is New Devices, A New Fan and the speaker will be Benjamin Adams.

Topics of discussion for the December 13 meeting have been chosen and are A New Steam Vacuum Pump and Metal Surfaces as Insulation. The speakers will be named at a later date.

The activities of the Philadelphia Chapter during the past year have been summarized in the annual report of Secy. W. R. Eichberg and eight meetings were held under the direction of W. F. Smith, chairman of the Meetings Committee.

October 12, 1933—A meeting was held at the Engineers Club, at which J. C. Albright presented a paper on Atmospheric Water Cooling Equipment. Attendance—60.

November 9, 1933—Meeting at the Engineers Club for dinner, followed by inspection of WCAU Broadcasting Studio, attended by 65 members and guests.

December 14, 1933—Meeting was held at the Engineers Club with 46 members and guests present. F. D. Berkeley spoke on Steam Jet Refrigeration.

January 16, 1934—The annual meeting was held at the Colonial Recreation Center, where election of officers for 1934 was held, followed by bowling and other sports, which was enjoyed by 101 members and guests.

February 8, 1934—Meeting was held at the Engineers Club and a paper on Hot Water Generators and the Btu as the Dollar Mark of Savings was presented to 33 members and guests by W. C. Beekley.

March 8, 1934—An inspection was made of the air conditioning system in the Philadelphia Savings Fund Society Building, where Ernest Dambly, representing the consulting engineer, and P. L. Davidson, representing the manufacturer, explained in detail the system installed in this 30 story building.

April 12, 1934—Fifty-three members and guests enjoyed dinner at the Engineers Club, and addresses by Ira Thomas on Amusing Experiences in Baseball, and V. P. Alley on The Swing to Purchased Steam in Philadelphia.

May 10, 1934—Annual outing meeting at the Roxborough Country Club, where golf, bridge and other amusements were participated in by 76 members and guests.

The membership list of the Philadelphia Chapter contains 125 with 85 members paid up.

The officers and committees will be in charge until the date of the annual meeting in January, with the exception of the Meetings Committee which will remain in office until May, 1935:

President—W. P. Culbert.
Vice-President—J. H. Hucker.
Secretary—W. R. Eichberg.
Treasurer—W. F. Smith.
Board of Governors—M. F. Blankin, A. C. Caldwell, Alexander McClintock, Jr.

COMMITTEES

Meetings
A. C. Caldwell, *Chairman*
S. E. Plewes
Morris Sheffler
C. B. Eastman
L. P. Hynes

Attendance
A. H. MacDade, *Chairman*
R. F. Hunger
J. L. Clodfelter

Finance
J. D. Cassell, *Chairman*
W. F. Smith
E. N. Sanbern
M. F. Blankin

Legislative
R. C. Bolsinger, *Chairman* (*1 year*)
L. C. Davidson (*2 years*)
F. D. Mensing (*3 years*)

Publicity
H. P. Gant, *Chairman*
H. H. Erickson
R. C. Bolsinger

Membership
Alexander McClintock, Jr., *Chairman*
Karl Rugart
H. F. Rettew
J. D. Cassell
A. E. Kriebel
H. G. Black

St. Louis

The St. Louis Chapter reports that it has had a very active year with many good meetings and speakers. In addition to the usual activities, the Chapter sponsored a series of meetings at the David Ranken, Jr. School of Mechanical Trades, comprising the Open Forum on Air Conditioning at which many of the speakers were members of the Chapter. The meetings were open to all and were exceptionally well attended, an indication of the present interest in air conditioning. On several occasions speakers were also furnished for luncheon clubs.

Secy. A. L. Walters reports that finances of the Chapter are in good condition with collections within $10.00 of the current budget and the prospect of actual expenditures being slightly less than anticipated.

The Entertainment Committee arranged a bridge party last spring which was enjoyed by many of the members and guests.

On November 8, 1934, a joint meeting with the Engineers' Club will be addressed by Pres. A. C. Willard, University of Illinois, and a past president of the Society, on Air Conditioning for Human Comfort. The annual meeting, election and installa-

tion of officers will take place on December 5, 1934, which will also feature ladies night and an entertainment.

At the present time the following officers and committee members are in charge:

President—Paul Sodemann
First Vice-President—J. W. Cooper.
Second Vice-President—C. R. Davis
Secretary—A. L. Walters
Treasurer—G. W. F. Myers
Board of Governors—C. A. Pickett, J. M. Foster, E. E. Carlson, E. B. Evleth

COMMITTEES

Program
J. W. Cooper, *Chairman*
E. E. Carlson
H. W. McLarney
R. J. Tenkonohy

Publicity
G. W. F. Myers, *Chairman*
G. L. Ranck
L. R. Szombathy

Entertainment
C. R. Davis, *Chairman*
R. R. Clegg
J. E. Hamilton

Membership
D. J. Fagin, *Chairman*
E. B. Evleth
C. E. Hartwein

Employment Officer
E. A. White

Educational
C. A. Pickett, *Chairman*
E. P. Bradley
G. B. Rodenheiser
R. M. Rosebrough
D. L. Moodie

Resolutions
W. C. B. Sodemann, *Chairman*
J. D. Falvey
H. A. Grossmann

Assoc. Engrg. Soc. Com.
Paul Sodemann, *Chairman*
L. W. Moon, *Counselor*
J. M. Foster, *Counselor*
C. A. Pickett, *Alternate*

Auditing
J. W. Cooper, *Chairman*
L. W. Moon
E. E. Carlson

A resumé of meetings held during the past season has been prepared:

October 4, 1933. Kingsway Hotel. Pres. W. T. Jones, Boston, Mass., spoke on The Status Quo of the Society and presented a very enlightening and detailed account of the activities of the A. S. H. V. E.. Attendance—29.

November 8, 1933. David Ranken School. F. D. Berkeley, Buffalo, N. Y., spoke on Decalorators—Steam Vacuum refrigeration and described the invention and evolution of steam vacuum refrigeration with the assistance of slides. Attendance—65.

December 6, 1933. Kingsway Hotel. Annual meeting, election and installation of officers. Heating and ventilating story telling contest. Attendance—30.

The first three meetings in 1934 on January 15, February 5, and March 5 were held at the David Ranken School and the dates coincided with meetings of the Open Forum on Air Conditioning, which the Chapter attended in a body after a dinner and business session. The attendance at the preliminary meetings was 25, 23 and 22.

April 4, 1934. Kingsway Hotel. W. P. Hacker, of Geo. E. Wells & Co., consulting engineers, gave a very enlightening and entertaining account of Experiences of an Engineer in China, describing his three year stay there doing engineering. Attendance—15.

May 9, 1934. Joe Garavelli's. Pres. C. V. Haynes, Philadelphia, Pa., explained in detail the conditions and affairs of the A. S. H. V. E. in his talk on Your Society—What Does It Mean to You. John Howatt, 1st vice-president and chairman of the Committee on Research, spoke on Research Activities of the Society and outlined the programs under way, as well as those contemplated. F. H. Gaylord related in a lighter vein The Experiences of a Manager. Attendance—25.

The number of members on the roll is 31 and during the past year five new members were added and two were reinstated.

Southern California

The Southern California Chapter planned an interesting program for the September meeting when arrangements were made to have the local administrator of the Federal Housing Commission as guest speaker. An explanation was given of the requirements necessary for building owners to obtain funds from the government at low interest rate and long term notes in order to modernize mechanical equipment of buildings.

The annual report of the secretary, E. H. Kendall, shows a total attendance of 249 members and 84 guests at the meetings during 1933-34, at which the following programs were presented:

Application of Air Conditioning to Greenhouses—Dr. L. C. Marshall, U. S. Dept. of Agriculture, Bureau of Plant Industry. Dr. Marshall has devoted considerable time to this subject and presented a very interesting and worthwhile paper. Four greenhouses in Southern California are completely air conditioned and show splendid results.

Steam Refrigeration and Its Application to Air Conditioning—C. S. Anderson. This meeting was well attended and attested to the interest of the subject.

Times Newspaper Building—E. L. Ellingwood. The problems encountered in the design of this structure, recently erected and completely air conditioned, were described by Mr. Ellingwood, consulting engineer.

The Whys and Wherefores of Weather—A. H. Daingerfield, director of the U. S. Weather Bureau.

Back Siphonage—O. W. Ott. Some very enlightening highlights were given on what sometimes occurs with cross-connections.

The officers who conducted the affairs of the Southern California Chapter during the past season are:

President—W. H. C. Ness.
Vice-President—W. E. Barnum.
Chairman—L. H. Polderman.
Secretary—E. H. Kendall.
Treasurer—H. L. Warren.
Board of Governors—W. E. Cranston, H. H. Douglas, A. H. Simonds.

Western New York

The secretary of the Western New York Chapter, P. S. Hedley, has outlined plans for future meetings. In November the speaker will be A. J. Nesbitt, Philadelphia, Pa., and in December the usual Christmas Bowling Party will be enjoyed.

The Chapter officers and committees are as follows:

President—J. J. Yager
Vice-President—W. E. Voisinet
Secretary—P. S. Hedley
Treasurer—B. C. Candee
Board of Governors—R. T. Coe, L. A. Harding, C. W. Farrar, W. G. Fraser, Roswell Farnham, M. S. Jackson, C. A. Evans, O. K. Dyer, Joseph Davis, M. C. Beman and D. J. Mahoney

COMMITTEES

Program
D. J. Mahoney, *Chairman*
M. C. Beman
F. H. Sharp

Entertainment
W. E. Voisinet, *Chairman*
Joseph Davis
Roger Thornton

Membership
Harry Hexamer
A. R. Acheson

A synopsis of meetings held during the past year is given:

January 8, 1934—Speaker. Mr. Burnham. Topic: Copper Pipe and Fittings.

February 28, 1934—Dinner dance and reception for officers.

March 12, 1934—Speaker: F. I. Raymond. Topic: Principles of Steam Conservation.

April 9, 1934—Speaker: L. A. Harding. Topic: Refrigeration.

May 14, 1934—Speakers: Pres. C. V. Haynes on National Society Activities and John Howatt, 1st Vice-President, on Research.

June 13, 1934—Golf meeting.

The average attendance at chapter meetings was 35 and 109 are listed on the Chapter membership roll.

Western Michigan

The annual report of the Western Michigan Chapter prepared by the secretary, W. G. Schlichting, gives a synopsis of meetings held during the past season:

October 9, 1933—R. J. Thompson presented The Refrigerant

Freon—A Scientific Development. Attendance—13 members and 10 guests.

November 14, 1933—M. A. Powers and E. E. Van Norstand presented Boiler Burner Units. Attendance—14 members and 26 guests.

December 11, 1933—Meeting in Kalamazoo. S. H. Downs demonstrated and discussed new air handling and conditioning equipment developments. Attendance—16 members and 48 guests.

January 8, 1934—H. E. Paetz presented The Decalorator. Attendance—11 members and 15 guests.

February 12, 1934—L. B. Miller presented Effective Temperature Control. Attendance—13 members and 16 guests.

March 12, 1934—Meeting in Lansing. Discussion of the papers, Thermodynamic Properties of Moist Air, and, The Comfort Zone and Factors Affecting It, by Dean H. B. Dirks and Prof. L. G. Miller. Attendance—15 members.

April 9, 1934—V. D. Milliken presented Vacuum Pumps. Attendance—13 members.

June 8, 1934—Annual meeting at Green Ridge Country Club, with golf and baseball during afternoon. Dinner was served in the club house, after which the following officers for the 1934-1935 season were elected:

President—S. H. Downs.
Vice-President—P. O. Wierenga.
Secretary—W. G. Schlichting.
Treasurer—W. J. Temple.
Board of Governors—K. L. Ziesse, H. F. Reid, Harry Christenson.

At the present time 25 members are on the Chapter's roll.

Death of Alexander McClintock, Sr.

On September 20, 1934 the heating industry and members of the Philadelphia Chapter in particular suffered a great loss in the death of Alexander McClintock, Sr.

A native of Ireland, where he was born in March 1856 at St. Johnson, County Donegal, Mr. McClintock migrated to this country at the age of 17 with his parents and sisters. He had the benefit of a short grammar school education and any deficiency in schooling was largely compensated for by natural acumen, energy and industry.

In Philadelphia Mr. McClintock was first employed by the late Dr. Filbert, a pioneer in the laying of asphalt pavements, and was in charge of work on the pavements of the Centennial grounds in 1876. His next position was with the James P. Wood Heating Co., where he learned the steamfitting trade, later becoming a superintendent for the Philadelphia Steam Heating Co. In 1900 he formed a partnership with J. W. Cuff and operated under the name of J. W. Cuff & Co., heating and ventilating engineers. This activity was successful from the start and continued until 1905, when Mr. McClintock started in business under his own name. In 1919 with his three sons he established the firm of A. McClintock & Sons, which is still in existence.

At the time of his death, in his 78th year, Mr. McClintock was a vestryman of the Reformed Episcopal Church of Our Redeemer, Philadelphia. He was a member of the *Philadelphia Heating and Piping Contractors Association* and until shortly before his death had been an active member of the Philadelphia Chapter of the Society, which he served as treasurer from 1923 to 1925.

Mr. McClintock's fraternal connections were extensive and of long standing. He became attached to the Order of Orangemen at the age of 14 and continued in the Order for 65 years, serving as Supreme Grand Master of the Loyal Orange Institution of the United States and as Supreme Grand Secretary. He was a Past Master of the Loyal Orange Lodge No. 59 of Philadelphia, member of Royal Arch Chapter Black Preceptory No. 34, president of the Independent Patriotic League of Philadelphia, director of the Orange Home at Hatboro, Pa., member of the Wm. L. Elkins Lodge No. 646 F. & A. M., member of St. Johns Royal Arch Chapter No. 32, and a member of Philadelphia Consistory.

Surviving Mr. McClintock are three sons, John L., Thomas A., and Alexander, Jr., and three daughters, Mrs. S. A. Clark, Mrs. A. H. King, and Mrs. S. Chandler. Interment took place September 24, in Fernwood Cemetery with impressive ceremonies conducted by the Masonic and Orange Orders.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 23 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by November 15, 1934, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	*REFERENCES*	
	Proposers	*Seconders*
BRINKER, H. A., Member of Firm, Wilson-Brinker Co., Kalamazoo, Mich.	S. H. Downs	R. R. Ware *(Non-Member)*
	C. R. McConner	W. G. Schlichting
DAVIS, A. F., Vice-Pres., The Johnson and Davis Plbg. & Htg. Co., Denver, Colo.	O. G. Ward	H. W. Ellis
	F. C. McIntosh	W. H. Wilson
DISNEY, M. A., Co-Partner, Disney-Leffel Co., Kansas City, Mo.	B. F. Cook	L. R. Chase
	E. K. Campbell	W. E. Gillham
EPPLE, A. B., Student Engr., B. F. Sturtevant Co., Hyde Park, Mass.	G. L. Larson	G. C. Wilson *(Non-Member)*
	D. W. Nelson	E. T. Hansen *(Non-Member)*